PERFORMANCE OPTIMISATION OF A COMPRESSION IGNITION ENGINE FUELLED ON ETHANOL

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A research report submitted to the Faculty of Engineering and the Built Environment, University of the Witwatersrand, in partial fulfilment of the requirements for the degree of Master of Science in Engineering

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DECLARATION

I declare that this research report is my own, unaided work. It is being submitted for the degree of Master of Science (Engineering) in the University of the Witwatersrand, Johannesburg. It has not been submitted before for any degree or examination in any other University.

_____ _____

Heinrich Richardt Teise Date

To my parents,

Hans and Caroline Teise

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ABSTRACT

In this research project, the performance and emissions of a conventional compression ignition engine fuelled on ethanol as main fuel and dimethyl ether as ignition promoter were investigated. Tests were first conducted on diesel fuel, then on ethanol fuel with dimethyl ether and compared. All tests for both fuelling techniques were conducted at the same engine speed and injector pressure. However, engine settings with specific reference to injection timing and injector pressure were optimised to suit diesel fuel, and were left unaltered when the engine was fuelled on ethanol and dimethyl ether. The injector nozzle configuration used for diesel fuel was a standard three-hole type nozzle, whereas for ethanol fuel with dimethyl ether a standard three-hole nozzle as well as a four-hole type nozzle was used. Also investigated was the effect a catalytic converter would have on exhaust emissions, from both fuelling techniques.

The performance results of ethanol/dimethyl ether fuel compared favourably to that of diesel fuel. The brake power attained for both fuelling techniques was approximately the same, however the only penalty incurred to this desired result was the simultaneous increase in the brake specific fuel consumption of ethanol/dimethyl ether fuel. The fuel conversion efficiency of ethanol/dimethyl ether fuel was also found to be lower than that of diesel fuel, this largely attributed to the difference in energy release patterns between the two fuels.

The emissions results obtained showed that ethanol/dimethyl ether fuel burns cleaner, mainly due to its chemical structure containing oxygen molecules. The NO_x, THC, CO and CO_2 emissions, produced before the catalytic converter, of ethanol/dimethyl ether fuel were lower than those of diesel fuel. The catalytic converter further produced lower emissions, with the four-hole type nozzle producing the most desired results. In terms of catalytic converter efficiency, THC and CO emissions were more readily removed compared to NO_x. In addition, virtually no smoke emissions were detected for ethanol/dimethyl ether fuel combustion.

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NOMENCLATURE

ABDC	After Bottom Dead Centre
ADC	Analogue-to-digital
ATC / ATDC	After Top Dead Centre
BBDC	Before Bottom Dead Centre
bmep	Brake mean effective pressure
bsfc	Brake specific fuel consumption
BTC / BTDC	Before Top Dead Centre
CA^{o}	Crank Angle Degrees
CLD	Chemiluminescence Detector
CNG	Compressed Natural Gas
DI	Direct Injection
DMC	Dimethyl Carbonate
DME	Dimethyl Ether
EGR	Exhaust Gas Recirculation
FID	Flame Ionisation Detector
FIE	Fuel Injection Equipment
hp	Horse power
HSU	Hartridge Smoke Units
IDI	Indirect Injection
LPG	Liquefied Petroleum Gas
MUR	Maximum Useful Rate
NDIR	Non-dispersive infra-red
RON	Research Octane Number
TC / TDC	Top Dead Centre
THC	Total Hydrocarbons
UHC	Unburned Hydrocarbons

Chemical Symbols and Formulae

CO	Carbon Monoxide
CO_2	Carbon Dioxide
CFC	Chlorofluorocarbon
CH_4	Methane
CH_3OH	Methanol
CH_3OCH_3	Dimethyl Ether
C_2H_6	Ethane
C_2H_5OH	Ethanol
C_3H_6	Cyclopropane
C_3H_8	Propane
C_4H_{10}	Butane
$CuSO_4$	Copper Sulphate
H	Hydrogen
HC	Hydrocarbon
H_2O	Water
N	Nitrogen
NO	Nitric Oxide
NO_x	Nitrogen Oxide
O	Oxygen
OH	Hydroxyl
Pb	Palladium
Pt	Platinum
Rh	Rhodium
SO_2	Sulphur Dioxide

1 INTRODUCTION

Mankind has always engineered ways to make life easier and more comfortable, and as a result transportation is one area where great success has been achieved. Transportation, as we have come to know, by land, air or sea, plays an essential role in our daily lives. To keep up with this role, the engine has evolved substantially over the years, from humble beginnings of practical heat engines that have served mankind for over two and a half centuries, to gas turbines that provide thrust for aeroplanes and generate electricity. For the first 150 years, water, raised to steam, was interposed between the combustion gases produced by burning the fuel and the work-producing piston-in-cylinder expander. However, it was not until after the 1860's when Otto first developed the spark-ignition engine and 1893 when Diesel invented the compression-ignition engine, that engine development progressed significantly.

These engines have continued to evolve as our knowledge of engines has increased, as new technologies became available, as demand for new types of engines arose, and as environmental constraints on engine use changed.

Internal combustion engines, and the industries that develop and manufacture them support their use, now play a dominant role in the fields of power, propulsion and energy. With the last three decades having an explosive growth in research and development as the issues of air pollution, fuel cost and market competitiveness have become ever increasingly important. The internal combustion engine's purpose is the production of mechanical power from the chemical energy contained in the fuel. However, the by-products of the combustion of the fuel have been the main concern, as it affects performance and most of all, the environment in the form of air pollution. The automotive air pollution problem initially became apparent in the 1940's in the Los Angeles basin. Smog was later identified as the problem, because of the visibility of smog, but was only part of the problem. Further problems and effects such as airborne lead, acid rain, global warming and ozone depletion were also identified.[1]

The harmful effects of airborne lead, particularly on the health of children, have been known for many years and the pressure to eliminate the use of lead alkyls, a particularly poisonous form of the element, as an octane number improver for petrol has succeeded in effectively banning the production of leaded fuels in most parts of the world. A further incentive to the use of unleaded fuels is that a very small amount of lead will permanently 'poison' a catalytic converter.

Since the early 1970's, acid rain, with its harmful effects on trees and other vegetation, has attracted increasing public attention. The mechanisms by which acid rain is formed involve chemical reactions that take place high in the atmosphere. The major acid precursors emitted as a result of man's activities are sulphur dioxide (SO_2) and nitric oxide (NO).

Photochemical smog, as mentioned before was first recognised in Los Angeles, is particularly apparent in cities where a small amount of coal is burned, there is minimal industrial activity and there are large concentrations of automobiles.[2] Photochemical smog results from reactions between oxides of nitrogen and hydrocarbon compounds in the presence of sunlight.[3]

Global warming is another matter of general public concern. A fact well known is that the existence of life on earth depends on the ability of certain trace gases in the atmosphere to absorb and re-emit a substantial fraction of the infra-red radiation, which the surface emits in response to solar heating and which in the absence of these gases, would escape to space. Water vapour is the most important natural greenhouse gas, but its concentration depends on the climate itself and is unaffected by human activities. The significant greenhouse gases that are affected by human activities are carbon dioxide (CO_2), methane (CH_4), nitric oxide (NO) and chlorofluorocarbons (CFC's).[2]

The stratospheric ozone layer absorbs much of the sun's ultra violet radiation and any depletion of this ozone could have serious effects on human and animal life and on vegetation. CFC's represent the most serious threat to the ozone layer, however not withstanding hydrocarbons and oxides of nitrogen, which are also harmful.[2]

With the above-mentioned repercussions, emissions standards in the US, Europe and Japan, and for other engine applications, have followed. Substantial reductions in emissions from spark ignition and diesel engines have been achieved. Both the use of catalysts in spark ignition engine exhaust systems for emissions control and concern over the toxicity of lead anti-knock additives have resulted in the reappearance of unleaded petrol as a major part of the automobile fuels market. Also, the maximum lead content in leaded petrol has been substantially reduced. The emission-control requirements and these fuel developments have produced significant changes in the way internal combustion engines are designed and operated.[1]

In response to the ever-tightening standards for exhaust emissions and the problem of fuel shortage with the ever-increasing number of automobiles, the search for an alternative fuel with high efficiency and low emissions for the automotive engine is becoming one of the key directions of engine development.[4]

The concept of using alcohol fuels as alternatives to diesel has been experimented for a number of years. The scarcity of transportation petroleum fuels, which developed in the early 1970's, spurred many efforts to find alternatives. Alcohols were quickly recognised as prime candidates to displace or replace high-octane petroleum fuels. However, alternatives to the large demand for diesel fuel in many countries were not as evident. Innovative thinking led to various techniques by which alcohol fuels could partially or completely displace diesel fuel in diesel transportation vehicles.[5]

In order to expand the research on the use of alcohol fuels, and since considerable research had been carried out on methanol and dimethyl ether, it was decided to study the possibility of using ethanol as the main fuel, in place of methanol. Extensive research has been conducted at the University of the Witwatersrand on methanol, as an alternative fuel, and has been compared with similar tests conducted on diesel fuel. However, less research has been conducted using ethanol, as an alternative fuel to compare it with methanol. Therefore the aim of this research was to conduct tests using ethanol as the main fuel and compare emissions and performance parameters with those of diesel fuel.

In addition to emissions analysis and comparison, the effects of a catalytic converter was investigated. Thus the broader scope of the current research was to observe the effect the catalytic converter would have on the emissions produced.

Possible thought may render that after a century of development, the internal combustion engine has reached its peak and minimal potential for further improvements exists. However, this is not the case. Conventional spark-ignition and diesel engines continue to show substantial improvements in efficiency, power and degree of emission control. The engine development opportunities of the future are substantial, and the research, which was conducted, is merely a small part of a very vast area of research.[1]

1.1 Atmospheric Pollution: Contributions more specifically made by the internal combustion engine

The internal combustion engine is involved in many types of pollution creating mechanisms viz, lead deposition, carcinogens, acid rain, global warming, smog and ozone depletion. Both the spark and compression ignition engines contribute substantially to atmospheric pollution, as shown in table 1.1 below, as compared to other pollution creating processes. Typical USA figures are shown.

Table 1.1: Gas contents from various sources[1] [3]

Source	CO (%)	Sulphur Oxides (%)	Hydrocarbons (%)	Particulates (%)	NO (%)
Transport	92	4	65	14	42
Industry	4	32	26	44	21
Generation of Electricity	0	48	0	21	32
Space Heating	3	12	3	14	5
Refuse Burning	1	4	6	7	0

As can be seen from the table, transportation is responsible for the biggest share of CO, HC, and NO in the atmosphere as well as a large proportion of particulate matter.

1.1.1 Emissions from Diesel Engines

The combustion of the diesel engine always operates with considerable excess air, so that CO emissions are not a significant problem and hence the close control of air-fuel ratio is not required. The air-fuel ratio in petrol engines determines the effect on the emission of the main pollutants: CO, NO_x and unburned hydrocarbons. However in diesel engines, particulates are much more of a problem, and NO_x production is substantial.[2]

[1] Particulates in the table do not include dust from the road, rubber plastics from the tyres, photochemical smog particles or asbestos from brake linings. They are merely the particles, mainly carbon, directly attributable to the exhaust systems.

The indirect injection (pre-chamber) engine performs well in terms of NO_x emissions, however the fuel consumption penalty associated with indirect injection has resulted in a general move to direct injection, associated with four valve cylinder heads with a central fuel injector. This development is associated with a sharp increase in fuel injection pressures, now commonly 1500 bar or more, and the shaping of the fuel injection pulse.[2]

The 'common rail' fuel injection system, where fuel is supplied at a uniform high pressure and injector needle movement controlled electrically, is making headway but carries a cost penalty. NO_x emissions are very sensitive to maximum cylinder temperature and to the excess air factor. This has prompted the use of increased levels of turbocharging with improved after-cooling, as well as the use of retarded injection which results in reduced peak pressures and temperatures. However beyond a certain point, increased fuel consumption occurs. Exhaust gas recirculation can also play a part, while in large marine engines the addition of up to 30% water to heavy fuel oils can reduce emissions substantially.[2]

Diesel particulate emissions, while greatly reduced since the days when heavy goods vehicles could be recognised by their black smoke, remain a problem. Developments in combustion chamber design and in fuel injection systems are helping, while attention to factors such as the reduction of lubricating oil consumption, the reduction of top land clearance and increased piston crown temperatures all make a contribution. However, the evolution of effective exhaust after-treatment system is not yet complete. A further emissions problem, for which engines cannot be blamed, concerns the presence of sulphur dioxide, SO_2, in the exhaust. Permitted levels of sulphur in fuels for road vehicles have been drastically reduced, and have given rise to incidental problems with fuel injection equipment arising from the reduced lubricity of the fuel.[2]

1.1.2 Nitrogen Oxides

Nitrogen oxides (NO_x) are formed throughout the combustion chamber during the combustion process due to the reaction of atomic oxygen and nitrogen. The reactions forming NO_x are very temperature dependent, so the NO_x emissions from an engine scale proportionally to the engine load, and NO_x emissions are relatively low during engine start and warm up.[6]

Three reaction mechanisms that produce NO are the thermal or Zeldovich mechanism, the Fenimore or prompt mechanism and the N_2O intermediate mechanism. For the internal combustion engines, the most significant is the Zeldovich mechanism (Zeldovich, 1946)[6] in which NO is formed in the high temperature burned gases left behind by the flame front.[6]

The prompt mechanism occurs within the flame front, and is relatively small if the volume of the high temperature burned gases is much larger than the instantaneous volume of the flame front, as is usually the case in internal combustion engines.[6]

The following three chemical equations form the extended Zeldovich reaction (Miller and Bowman, 1989)[6]:

$$O + N_2 \leftrightarrow NO + N$$

$$N + O_2 \leftrightarrow NO + O$$

$$N + OH \leftrightarrow NO + H$$

All NO_x compounds are destroyed only very slowly at room temperature and remain in the atmosphere for considerable time. The human effects include toxicity, which occurs because the haemoglobin in the blood has a higher affinity for NO than for oxygen. The NO then has the ability to form dilute nitric acid in the lungs, which would result in breathing problems. [3,6]

1.1.3 Carbon Monoxide

Carbon Monoxide (CO) appears in the exhaust of rich-running engines since there is insufficient oxygen to convert all the carbon in the fuel to carbon dioxide. The most important engine parameter influencing carbon monoxide emissions is the fuel-air ratio. All the other variables cause second order effects. Thus, results obtained when varying fuel-air ratio are more or less universal. Hence the key to minimising CO emissions is to minimise the times the engine runs rich (such as during start-up). Since diesel engines run lean overall, their emissions of carbon monoxide are low and generally not considered a problem. It does appear, from previous tests, that direct injection engines emit relatively more CO than indirect diesel engines. Toxicity occurs very much the same way as NO and causes suffocation.[6]

1.1.4 Carbon Dioxide

Carbon dioxide (CO_2) is a colourless and odourless gas, but unlike carbon monoxide, it consists of two oxygen atoms for every carbon atom. It forms 0.03% of the air naturally. It is produced by human activities during the combustion of hydrocarbons, including fossil fuels. Carbon dioxide does not directly impair human health, but it is the main greenhouse gas that traps the earth's heat and causes global warming. Although it has significant impact on the environment, it is not a major contributor to diesel emissions as compared to the other gases.[6]

1.1.5 Hydrocarbons

Hydrocarbons are a group of compounds consisting of carbon atoms and hydrogen atoms. Hydrocarbons from diesel engines come primarily from; (1) fuel trapped in the injector at the end of injection that later diffuses out, (2) fuel mixed into air surrounding the burning spray, so lean that it cannot burn, and (3) fuel trapped along the walls by crevices, deposits, or oil due to impingement by the spray. The diesel combustion process relies on mixing fuel and air at the same time they are intended to burn. Hydrocarbon emissions are prominent from both a naturally aspirated direct injection engine and a naturally aspirated indirect injection engine, however in general direct injection engines emit more hydrocarbons than indirect injection engines.[6] The amounts are small compared to CO but are objectionable because of the odour, their influence in the formation of photochemical smog and possible carcinogenic effects. The odour is more noticeable with extremely lean mixtures typical of diesel or stratified charge engines. Hydrocarbons may also show up as particulate matter and in older engines may contain lead and engine scavenging compounds.[3]

1.1.6 Particulates

A high concentration of particulate matter is manifested as visible smoke or soot in the exhaust gases. Particulate emissions from engines are regulated since inhalation of small particulate matter can create respiratory problems. Particulates are a major emissions problem for diesel engines, as their performance is smoke limited. With the use of unleaded fuel, particulates are generally not as serious a problem for spark ignition engines. Particulates are any substance other than water that can be collected by filtering the exhaust.

Specifically, the U.S. Environmental Protection Agency defines a particulate as any substance other than water that can be collected by filtering diluted exhaust at or below 325 K. The particulate material collected on a filter is generally classified into two components. One component is a solid carbon material or soot and the other component is an organic fraction consisting of hydrocarbons and their partial oxidation products that have been condensed onto the filter or absorbed to the soot. The organic fraction is influenced by the processes that dilute the exhaust with air upon expulsion from the engine.[6]

1.1.7 Photochemical Smog

Photochemical smog is a mixture of ozone, aldehydes, oxides of nitrogen and hydrocarbons. It results from the reaction of these compounds in the atmosphere via a complex chain mechanism, which requires photolysis due to the action of sunlight. The amount of photochemical smog depends on the concentration of reactants, their reactivity (unsaturated hydrocarbons are more reactive than saturated), the temperature and light intensity. The characteristic brown colour associated with photochemical smog is due to NO in the atmosphere. This smog causes severe irritation of the eyes, throat and respiratory system, as well as damage to some material and thus a major problem especially when it is held down in the local atmosphere of major cities by a temperature inversion.[3]

1.1.8 Emissions Legislation

Emission control legislation for vehicles began in California due to the severe smog problems in the Los Angeles basin where high automobile usage is combined with difficult geography (a basin surrounded by mountains) and climate factors (frequent temperature inversions).

The legislation originated from the USA, the UN Economic Commission for Europe (ECE) and the European Union (EU), together with national standards for 18 individual countries. The majority of this legislation is concerned with specifying limits for the main pollutants: carbon monoxide, unburned hydrocarbons, oxides of nitrogen and exhaust particulates. This legislation has been largely driven by the USA, the first country to set emissions standards for vehicles, and the standards set by the US Environmental Protection Agency (EPA), has been adopted by many other countries.

The generally less stringent standards set by the United Nations Economic Commission for Europe are used in the European Union and a number of other countries. Japan has adopted rather different standards and test procedures.

The table 1.2 below gives an indication of the present state of progress of the main American, ECE, and other standards.

Table 1.2: Vehicle Emissions Standards [2]

Source	CO	Hydrocarbons	NO_x	Particulates
USA grams per mile, gasoline (petrol) and diesel engines				
Light Duty Vehicles				
Pre-1968 (uncontrolled)	90	15	6.2	-
Tier 1, 1994 - 1996	3.4	0.25	0.4	0.08
Tier 2, 2004	1.7	0.125	0.2	-
Low emission vehicle (LEV) 10 – 14 000 lb	7.0	0.3	2.0	-
Ultra low emission vehicle (ULEV) 10 – 14 000 lb	3.5	0.18	1.0	-
Europe, grams per km				
Passenger cars, gasoline (petrol) engines				
Effective 2000	2.3	0.2	0.15	-
Effective 2005	1.0	0.1	0.08	-
Passenger cars, diesel engines				
	CO	**Hydrocarbons + NO_x**		**Particulates**
Effective 2000	0.64	0.56		0.05
Effective 2005	0.50	0.30		0.025

The subsequent table 1.3 shows the position for heavy-duty (road) diesel engines, also the fairly exacting standards that are to be set for European railway locomotives.

In the case of marine diesel engines, which in many cases are called upon to burn the heaviest and most polluted residual fuel, a start is being made in attempting to limit NO_x emissions to a reasonable level.[2]

Table 1.3: Diesel Emissions Standards [2]

Source	CO	Hydrocarbons	NO_x	Particulates
USA grams per bhp-hr				
California, heavy duty	15.5	1.3	5.0	0.10
Medium duty	14.4	1.3	3.5	0.10
Europe grams per kWh				
ECE 49 (13 mode)	14.0	3.5	18.0	-
Clean Lorry Directive				
Euro 3, 1999	2.5	0.7	5.0	< 0.12
European Railways (UIC)	3.0	0.8	12.0	(Bosch 1.6 – 2.5)
Marine (draft IMO-regulation)	-	-	10-17	-

1.2 Alternative Fuels

Due to the harmful emissions from diesel engines and legislative requirements, much research has been expended into the replacement of diesel fuel. The replacements, known as alternative fuels, not currently used in vehicular applications for both economic and engineering reasons, possess much lower emission levels than petrol and diesel, and can be refined more easily from renewable feedstock. Alternative fuels research is also being spearheaded by mounting pressures of unavailability of crude oil, due to worldwide geo-political problems. The alternative fuels are propane, natural gas, hydrogen, methanol, ethanol and dimethyl ether.[6]

1.2.1 Propane

Propane (C_3H_8) is a saturated paraffinic hydrocarbon. When blended with butane (C_4H_{10}) or ethane (C_2H_6) it is also designated as liquefied petroleum gas (LPG). Propane has been used as a vehicular fuel since the 1930's. In 1993, there were about 4 million LPG vehicles operating worldwide, with the majority in the Netherlands, followed by Italy, the US and Canada. There is a relatively extensive refuelling network for propane, with over 15 000 refuelling stations available in North America. There are a number of original equipment manufacturers that currently sell propane-fuelled vehicles, primarily light and medium duty fleet vehicles, such as pick-up trucks and vans. Conversion kits are also available to convert petrol or diesel fuelled engines to dedicated propane or dual fuel use. In vehicles, propane is stored as a compressed liquid, and has evaporative emissions that are essentially zero, since it is used in a system that is sealed. Propane has an octane number of 112 on the Research Octane Number (RON) scale, so vehicular applications of propane will generally raise the compression ratio. This results in lower HC and CO emissions than petrol and diesel fuels, however NO_x levels tend to be higher.[6]

1.2.2 Natural Gas

Natural gas is a naturally occurring fuel found in oil fields. It is primarily composed of approximately 90 to 95% methane (CH_4), with small amounts of additional compounds such as 0 to 4% nitrogen, 4% ethane, and 1 to 2% propane. Methane is a greenhouse gas, with a global warming potential approximately ten times that of carbon dioxide, however methane has a lower carbon to hydrogen ratio relative to petrol, so its CO_2 emissions are about 22 to 25% lower than petrol.

Natural gas has been used for many years in stationary engines for gas compression and electric power generation. An extensive distribution network of natural gas pipelines exists to meet the need for natural gas for industrial processes and heating applications. Natural gas fuelled vehicles have been in use since the 1950's, and conversion kits are available for both spark and compression ignition engines. Recent research and development work has included development of bifuel vehicles that can operate either with natural gas and petrol or diesel fuel. One advantage of a bifuel operation is that the operating range of a vehicle is extended in comparison with a dedicated natural gas vehicle. Currently, original equipment manufacturers are selling production natural gas fuelled vehicles, primarily to fleet owners. Natural gas vehicles were the first vehicles to meet the Californian ultra-low emission vehicle (ULEV) emissions standards. Natural gas has an octane number (RON) of about 127, such that natural gas engines can operate at a compression ratio of 11:1, greater than petrol engines. Natural gas can replace diesel fuel in heavy duty engines with the addition of a spark ignition system. A number of heavy duty diesel engine manufacturers are also producing natural gas heavy duty engines. The natural gas fuelled engines are operated lean with an equivalence ratio as low as 0.7. The resulting lower in-cylinder temperatures reduce the NO_x levels. Natural gas can also be used in compression ignition engines if diesel fuel is used as a pilot fuel, since the autoignition temperature of methane is 540ºC compared to 260ºC for diesel fuel. This fuelling strategy is attractive for heavy duty diesel applications, such as trucks, buses, locomotives, ships, compressors and generators. These engines are also operated with a lean combustion mixture, so that the NO_x emissions are decreased. However, since diesel engines are unthrottled, at low loads, the lean combustion conditions degrade the combustion process, increasing the hydrocarbon and carbon monoxide emissions.[6]

1.2.3 Hydrogen

Hydrogen (H_2) can be produced from many different feedstock, including natural gas, coal, biomass and water. The production processes include steam reforming of natural gas, presently the most economical method; electrolysis of water and gasification of coal, which also produces CO_2. Hydrogen is colourless, odourless and non-toxic, and hydrogen flames are invisible and smokeless. The global warming potential of hydrogen is insignificant in comparison to hydrocarbon based fuels since the combustion of hydrogen produces no carbon based compounds such as HC, CO and CO_2. At present the largest user of hydrogen fuel is the aerospace community for rocket fuel. Hydrogen can also be used as a fuel in fuel cells.

There have been a number of vehicular demonstration projects, but the relatively high cost of hydrogen fuel has hindered adoption as an alternative fuel.[6]

Dual fuel engines have been used with hydrogen, in which hydrogen is used at start up and low load, and petrol at full load to reduce the cold start emissions levels. One of the major obstacles related to the use of hydrogen fuel is the lack of any manufacturing, distribution and storage infrastructure. The most economical would be to distribute hydrogen through pipelines, similar to natural gas distribution.[6]

Compressed hydrogen at 70 MPa has one-third the energy density by volume of compressed natural gas, and liquid hydrogen has one-fourth the energy density by volume of petrol. Use of liquid hydrogen has an additional energy cost, as liquefaction of hydrogen to –20K requires an expenditure of energy approximately equal to the energy content of the liquid hydrogen. The octane rating of hydrogen of 106 RON allows use of an increased compression ratio. The combustion characteristics of hydrogen are very different from those of petrol. This difference can be exploited by means of exhaust gas recirculation (EGR) and lean operation mechanisms, resulting in significantly lower NO_x levels.[6]

1.2.4 Methanol

Methanol (CH_3OH) is an alcohol fuel formed from natural gas, coal or biomass feedstock. Methanol is also called “wood alcohol”. It is a liquid at ambient conditions and its chemical structure is a hydrocarbon molecule with a single hydroxyl (OH) radical. The hydroxyl radical increases the polarity of the hydrocarbon, so that methanol is miscible in water, and has a relatively low vapour pressure. Since oxygen is part of the chemical structure, less air is required for complete combustion. Methanol is toxic and ingestion can lead to blindness and death. Methanol has been used as a vehicular fuel since the early 1900’s, and is also used as a fuel for diesel engines and fuel cells. Pure methanol is labelled M100, and a mix of 85% methanol and 15% petrol or diesel is labelled M85. Adding petrol to methanol provides more volatile components that can vapourise more easily at low temperatures. Methanol has been adopted as a racing fuel, both for performance and safety reasons. Since methanol mixes with water, a methanol fire, can be extinguished with water, which is not the case with petrol. The cetane number of methanol is low at about 5, but it can be used in compression ignition engines, however with an ignition promoter. One such promoter is Dimethyl Ether (DME).

The CO_2 emissions of methanol on an equivalent energy basis are about 96% that of petrol, non-methane hydrocarbons and CO also decreases, however NO_x emissions are found to be similar.[6]

1.2.5 Ethanol

Ethanol (C_2H_5OH) is an alcohol fuel formed from the fermentation of sugar and grain stocks, primarily sugar cane and corn, which are renewable energy sources. Its properties and combustion characteristics are very similar to those of methanol. Ethanol is also called "grain" alcohol. It is a liquid at ambient conditions, and non-toxic at low concentrations. Gasohol[1] (E10) is a gasoline-ethanol blend, with about 10% ethanol by volume. E85 is a blend of 85% ethanol and 15% petrol. In Brazil, about half of the vehicles use an ethanol-based fuel colloquially dubbed "alcool", which is primarily E93, produced from sugar cane. In the United States, the primary source of ethanol is currently from starch feedstock, such as corn, and there are efforts underway to produce ethanol from cellulose feedstock such as corn fibre, forestry waste, poplar and switch grass. The energy density by volume of ethanol is relatively high for an alternative fuel, about two-thirds that of petrol. The octane rating of ethanol of 111 (RON) allows use of an increased compression ratio and it can be used in compression ignition engines with an ignition promoter as in the case of methanol. Previous research (Cadle et al., 1997)[6] shows CO_2 emissions on an equivalent energy basis are about 99% that of petrol; NO_x emissions are lower, however CO emissions are increased.[6] The research conducted in this report will consolidate those results and also reveal optimised diesel engine parameters for lower emissions and improved performance.

1.2.6 Dimethyl Ether (DME)

Dimethyl Ether (CH_3OCH_3) has attracted increasing attention in engine development worldwide owing to its excellent characteristics in reducing emissions. DME can be produced from either methanol or natural gas, or from coal at an acceptable cost with technology available. It will attract more interest in countries with abundant coal and natural gas resources, easing their fuel problems as well as helping them to tackle their pollution problems. DME is one of the simplest ether compounds, with a low heating value that is only 64.7% of diesel, and therefore a larger amount of fuel supply is needed to ensure equal engine power output.

[1] Gasoline, being the American term for petrol.

The cetane number of DME is higher and the autoignition temperature is lower than in the case of diesel, which ensures a shorter period of ignition delay and a lower combustion noise compared to diesel operation. Thus DME is used in diesel engine testing as an ignition promoter for alcohol based fuels. DME has only C-H bonds, C-O bonds and no C-C bonds. Moreover DME contains about 34.8% oxygen, and therefore combustion produced emissions such as CO, HC, CO_2, smoke and particulate matter are expected to be lower than those of diesel operation. In addition, DME will tolerate a higher exhaust gas recirculation ratio to reduce NO_x emissions. The latent heat of vapourisation of DME is much higher than that of diesel [6] and this will be beneficial to NO_x reduction owing to the larger temperature drop of the fuel mixture in the cylinder. DME is in the gaseous state even at -20ºC and ambient pressure, thus can be handled without hazard by people. However, DME should be pressurised in the fuel tank owing to its low boiling point.

2 OBJECTIVES

The objective of this project was to investigate the differences in performance and emissions of a compression ignition engine fuelled firstly on diesel fuel and then on ethanol/DME fuel.

This objective was achieved by investigating:

- Performance of the engine using standard 3 hole and 4 hole injectors, at a constant injector pressure of 210 bar, and at a speed of 1550 rpm.
- Settings for reduced emissions at optimal performance.
- Emissions of the engine before and after the catalytic converter.
- Valid and useful conclusions and recommendations from the test results.

3 LITERATURE SURVEY

3.1 The Compression Ignition Engine: A Brief History

In 1893, a German patent was granted to Rudolf Diesel for the design of a “rational heat engine”. Diesel planned a four-stroke engine, which would incorporate the constant temperature energy addition of the Carnot cycle. In his first engine, ammonia was injected into the cylinder at the end of the compression stroke to avoid premature ignition. Diesel’s second engine in 1896, had a water-jacketed cylinder and pump to supply air to the cylinder to reduce exhaust smoke. In 1898, the first production engine ran on kerosene and gave an amazing 20 hp. By 1901, the external crosshead arrangement used in the original design was replaced by a trunk piston design innovated by the American Diesel Company. The M.A.N Company built a 250 hp, two-cylinder engine in 1902, and the Sulzers Company built a three- cylinder engine of 300 hp in 1906. Because such engines could not be directly reversed, marine applications used an electric drive for slow speed and maneuvering. A steam engine with oil-fired boilers and a diesel engine were shown side by side at the Turin Exhibition of 1911. The steam engine used several times more fuel per horsepower than the diesel engine. This greatly improved fuel economy foretold the demise of steam engines.[7]

By 1910, German and British companies had developed diesel-powered submarines, and the first diesel-powered passenger ship appeared in 1921. Early four-stroke ship engines experienced considerable problems with fouling of the exhaust valves and ports with carbon. This problem led Sulzers to develop the two-stroke diesel engine. Engines with 2000 hp per cylinder were in operation before World War I, and by 1939 half of the world’s shipping tonnage was diesel powered. High-speed diesel engines for commercial, farm and industrial applications developed slowly because advances were needed in strength of materials and fuel injection systems. Although the principle of airless injection of liquid fuel was pioneered by Herbert Akroyd Stuart as early as 1886, it was not until 1936 that Robert Bosch introduced an ingenious method of metering which did not require a variable-stroke pump. Equally important, the Bosch Company had the ability to use the high-precision machining required to fabricate such systems.[7]

Although diesels have displaced spark ignition engines in nearly all applications except automobiles and aircraft, a detailed understanding of the diesel combustion process is yet to be attained. Thus highly different combustion chamber designs have persisted, such as, for example, open-chamber designs where the fuel is directly injected into the cylinder, and pre-chamber designs where the fuel is injected into a small chamber, which is attached to the cylinder by a small passage. The difficult requirements of fuel-air mixing are met either by high-pressure injection (1000 atm) or by lower pressure injection (300 atm) combined with swirl imparted to the cylinder air by the induction process. The modern diesel engine is a highly sophisticated device. It can produce high power with excellent fuel economy while still meeting strict gaseous emission requirements. The recent addition of exhaust particulate regulation has, however, raised doubts concerning the future of the diesel engine for automobile and other light-duty transportation applications. Resolution of these many, often conflicting demands will require an improved understanding of the chemical kinetics and fluid mechanics which govern diesel combustion and diesel engine development.[7]

3.2 Variables that affect CI Engine Performance, Efficiency and Emissions

3.2.1 Load and Speed

The performance of a naturally aspirated direct injection (DI) heavy-duty truck diesel engine and a small indirect injection (IDI) engine at full load over the engine speed is shown in the figure 3.1 below.

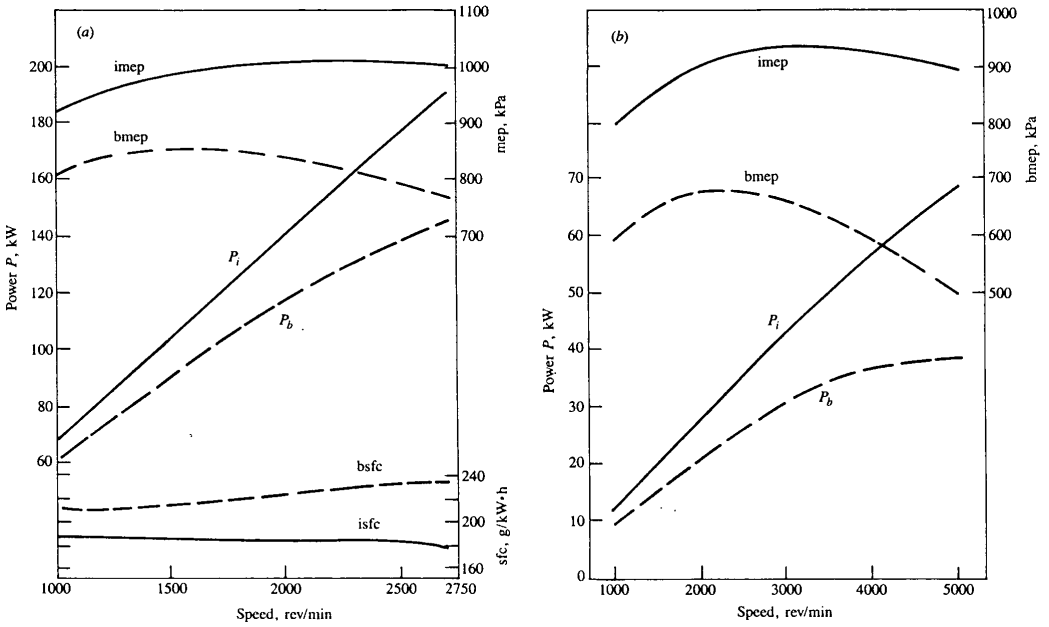

Figure 3.1: Gross indicated and brake power (P_i, P_b), mean effective pressure (imep, bmep), and specific fuel consumption (isfc, bsfc) for: (a) 8.4 dm^3 six-cylinder naturally aspirated direct injection engine; (b) 1.8 dm^3 four cylinder naturally aspirated indirect injection swirl-chamber diesel engine [1]

The figure shows the full load indicated and brake power, and mean effective pressure (mep) for naturally aspirated DI and IDI compression ignition engines. Except at high engine speeds, brake torque and mep vary only modestly with engine speed, since the intake system of the diesel can have larger flow areas than the intake of spark ignition engines. The part-load torque and brake mean effective pressure (bmep) characteristics (at fixed amount of fuel injected per cycle), have a similar shape to the full-load characteristics in the figure above.[1]

Performance maps where brake specific fuel consumption (bsfc) contours are plotted on a graph of bmep versus engine speed are commonly used to describe effects of load and speed variations. Figure 3.2 shows the performance map of an air-cooled four stroke cycle medium-swirl naturally aspirated DI engine.

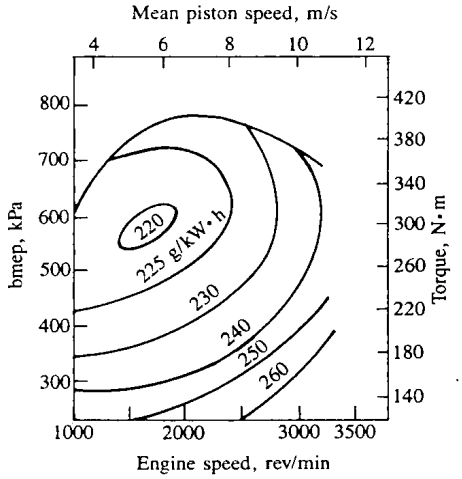

Figure 3.2: Performance map for 6.54 dm^3 eight-cylinder air-cooled naturally aspirated medium-swirl DI diesel engine. Contours of constant bsfc in grams per kilowatt-hour are shown. Multi-hole fuel nozzle [1]

Maximum rated power for this 6.54 dm^3 displacement engine at 3200 rpm is 119 kW, maximum bmep at 2000 rpm is 784 kPa, and minimum bsfc (at 1600 rpm and 580 kPa bmep) is 220 g/kW·h, which corresponds to a brake fuel conversion efficiency of 38.5%. The gross indicated fuel conversion efficiency would be about 48%.[1]

Figure 3.3 shows the performance map for a high-swirl DI diesel which uses a single fuel jet sprayed tangentially into the swirling air flow.

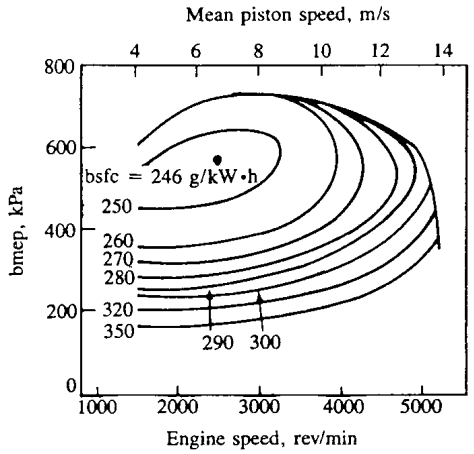

Figure 3.3: Performance map for 1.47 dm^3 four-cylinder naturally aspirated DI diesel engine with high-swirl single-hole-nozzle M.A.N. combustion system. Contours of constant bsfc in grams per kilowatt-hour shown [1]

Due to the higher speed and higher swirl than the larger DI engine in figure 3.2, the maximum bmep is slightly lower. The best bsfc is about 10 % higher largely due to higher friction mep, but in part due to higher heat losses resulting from the less favourable surface/volume ratio of the smaller bore engine and higher swirl, and lower heat-release rate of the small high-swirl DI engine. Note that the maximum mean piston speed for this engine, 13.3 m/s at 5000 rpm, is comparable to that of the larger medium-swirl engine in figure 3.2 (10.7 m/s). The increase in bsfc from the minimum value with increasing speed at constant load is due to the increase in friction mep, partly offset by the effect of decreasing importance of heat losses per cycle on efficiency. The increase in bsfc with decreasing load at constant speed is dominated by the decreasing mechanical efficiency as bmep is reduced. The indicated fuel conversion efficiency increase is partly offset as the fuel/air equivalence ratio is decreased. The trends in bsfc when increasing load at constant speed and increasing speed at constant load from the minimum are more modest. They are the net results of (1) the increase in mechanical efficiency and decrease in indicated fuel conversion efficiency as the load increases and (2) decreasing indicated fuel conversion efficiency due to increasing importance of heat losses and increasing mechanical efficiency as the speed decreases.[1]

Figure 3.4 below shows the effect of load on NO_x and HC emissions.

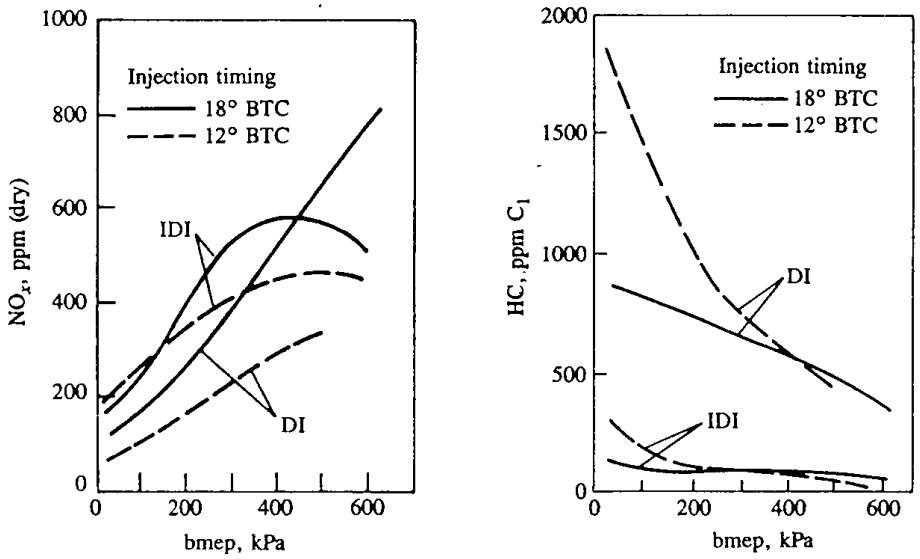

Figure 3.4: Effect of load on naturally aspirated diesel engine NO_x and HC emissions at rated speed, with two injection timings. Direct injection and indirect injection (pre-chamber) combustion systems. Six cylinder, 5.9 dm^3 displaced volume, engine [1]

For the DI engine, NO_x concentrations rise steadily as the fuel/air ratio increases with increasing bmep at constant injection timing. The increasing quantity of fuel injected per cycle results in an increasing amount of close-to-stoichiometric combustion products near the peak pressure and temperature, these characteristics do not change substantially with engine speed.[1]

3.2.2 Fuel injection Parameters

Fuel injection timing essentially controls the crank angle at which combustion starts. While the state of the air into which the fuel is injected changes as injection timing is varied, and thus ignition delay will vary, these effects are predictable. The fuel injection rate, fuel nozzle design (including number of holes), and fuel injection pressure all affect the characteristics of the diesel fuel spray and its mixing with air in the combustion chamber. Figure 3.5 overleaf, shows the effect on performance and emissions of varying injection timing, in (a) medium-swirl DI diesel engine and (b) an IDI engine. At fixed speed and constant fuel delivery per cycle, the DI engine shows an optimum bsfc and bmep at a specific start of ignition for a given injection duration.

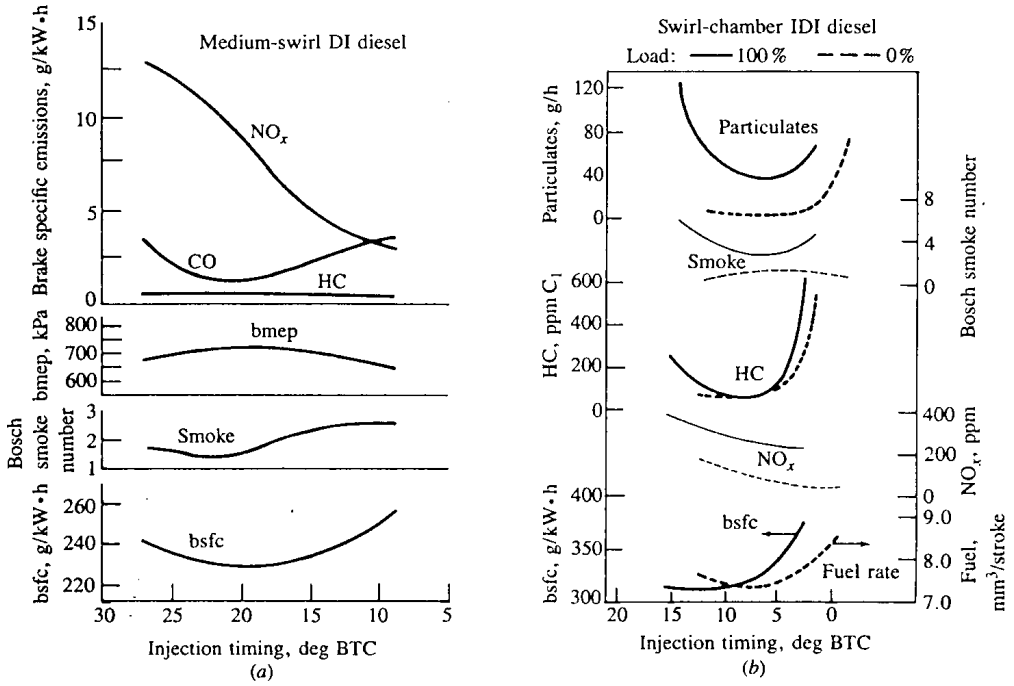

Figure 3.5: Effect of start-of-injection timing on diesel engine perfomance and emissions. (a) Medium-swirl DI diesel engine with deep combustion bowl and four-hole injection nozzle, 2600 rpm, fuel delivery 75 mm^3/cycle, fuel/air equivalence ratio 0.69. (b) Swirl-chamber IDI engine, 2500 rpm, 0 and 100 % load [1]

Injection timing variations have a strong effect on NO_x emissions for DI engines. Retarded injection is commonly used to help control NO_x emissions. It gives substantial reductions, initially with only a modest bsfc penalty. For the DI engine, at high loads, specific HC emissions are low and vary modestly with injection timing. For the IDI engine, at lighter loads, HC emissions are higher and increase as injection becomes significantly retarded from optimum. This trend is especially pronounced at idle.

Retarding timing generally increases smoke, though trends vary significantly between different types and designs of diesel engines. Mass particulate emissions increase as injection is retarded. The injection rate depends on the fuel injector nozzle area and injection pressure. Higher injection rates result in higher fuel-air mixing rates, and hence higher heat-release rates. For a given amount of fuel injected per cycle, as the injection rate is increased, the optimum injection timing moves closer to top dead centre crank position (TDC). The effects of injection rate and timing on bsfc in a naturally aspirated DI diesel engine are shown in figure 3.6.

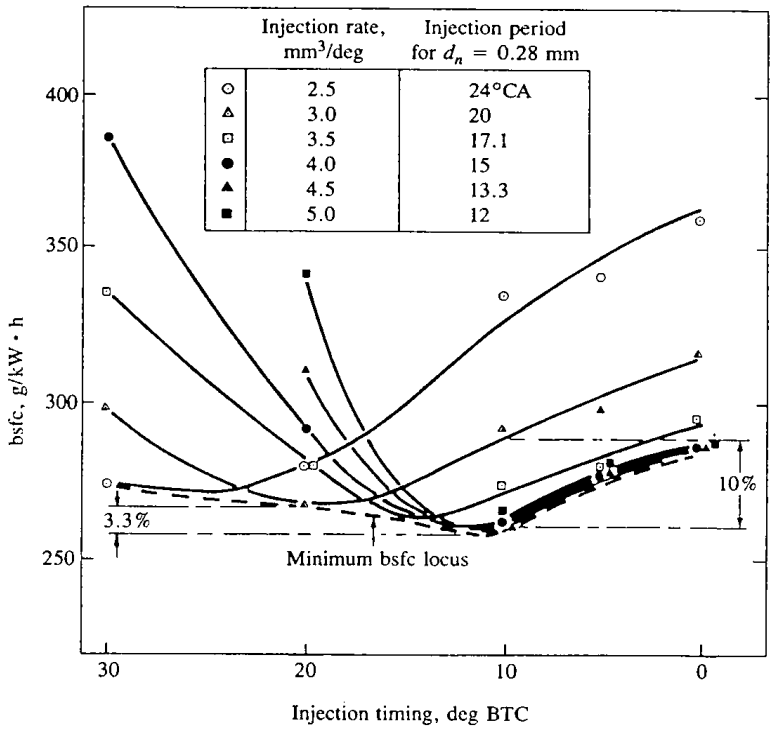

Figure 3.6: Effect of injection timing and injection rate on bsfc for 0.97 dm^3 single – cylinder naturally aspirated DI diesel engine with swirl. 2000 rpm, 60 mm^3 per stroke fuelling rate [1]

The higher the heat-release rates and shorter overall combustion process that result from the increased injection rate, decrease the minimum bsfc at optimum injection timing; however, a limit to these benefits is eventually reached. Increasing the injection rate increases the NO_x emissions and decreases smoke or particulate emissions. The controlling physical process is the rate of fuel-air mixing in the combustion chamber, so at constant fuel injected per cylinder per cycle, both increased injection pressure at fixed nozzle orifice area (which reduces injection duration) and reduced nozzle area at fixed injection duration, produce these trends.[1]

3.3 Catalytic Converter

The three-way catalytic converter is currently the most important exhaust gas treatment device, first installed on the exhaust systems in passenger cars in 1975. The name is derived from the fact that it works on all three of the gaseous pollutants of concern: nitric oxides, carbon monoxide and hydrocarbons. The operation of the catalytic converter is severely inhibited by lead and sulphur compounds in the exhaust gases, so that vehicular fuels have been reformulated to reduce their lead and sulphur content. All catalytic converters are built in a honeycomb or pellet geometry to expose the exhaust gases to a larger surface made of small particles (< 50nm) of one or more of the noble metals, platinum (Pt), palladium (Pb) and rhodium (Rh). Rhodium is the principal metal used to remove NO. Platinum is the principal metal used to remove HC and CO. The figure 3.7 below is a schematic of a three-way honeycomb catalyst.[6]

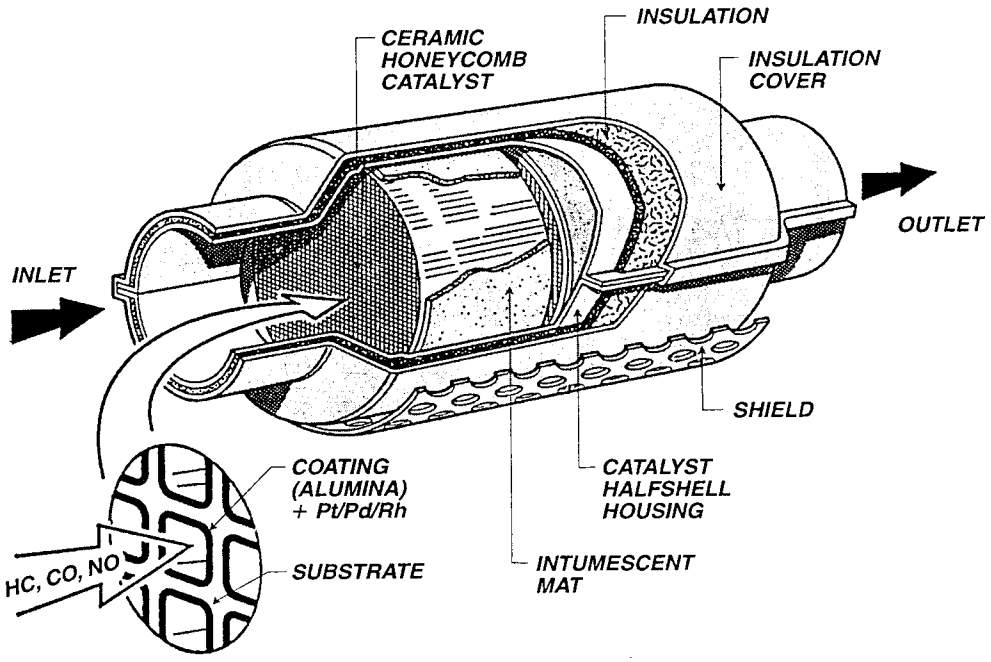

Figure 3.7: Catalytic Converter components. Courtesy Englehard Corporation [6]

In the converter shown, a thin layer of the noble metals covers a washcoat of inert alumina Al_2O_3 on a cordierite honeycomb foundation. As the exhaust gases flow through the catalyst, the NO reacts with the CO, hydrocarbons and H_2, via a reduction reaction on the surface of the catalyst. The remaining CO and hydrocarbons are removed through an oxidation reaction forming CO_2 and H_2O products.

The oxidation rate of hydrocarbons increases with molecular weight, such that the oxidation of low molecular weight fuels such as methane is very slow in the converter.[6]

A three-way catalyst will function correctly only if the exhaust gas composition corresponds to nearly (+/- 1%) stoichiometric combustion. If the exhaust is too lean, nitric oxides are not destroyed and if it is too rich, carbon monoxide and hydrocarbons are not destroyed. The figure below illustrates this phenomenon.[6]

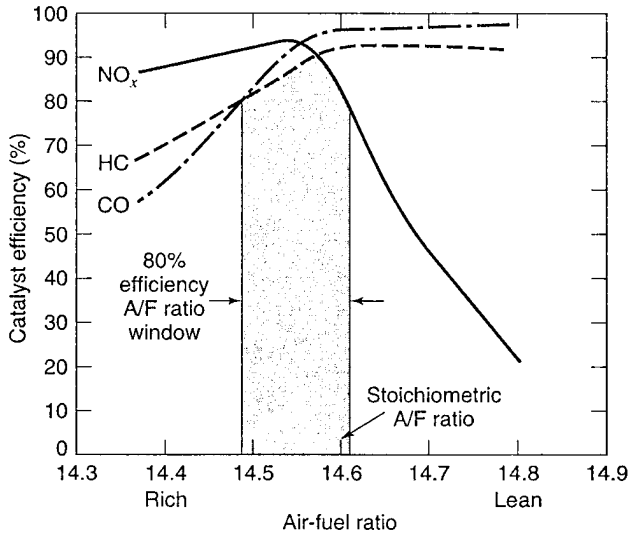

Figure 3.8: Conversion efficiency of a three-way catalyst as a function of air-fuel ratio [6]

Herein lies one constraint that emission control imposes upon engine operation; to use a three-way catalyst, the engine must operate in a narrow window about stoichiometric fuel-air ratios. A closed loop electronic control system with an oxygen sensor is used to determine the actual fuel-air ratio, and adjust the fuel injector so that the engine operates in a narrow range about the stoichiometric set point, this mechanism is a very costly one. Ordinary carburettors, are not able to maintain the fuel-air ratio in such a narrow set point range.[6]

The catalytic conversion efficiency is plotted versus temperature in figure 3.9 below.

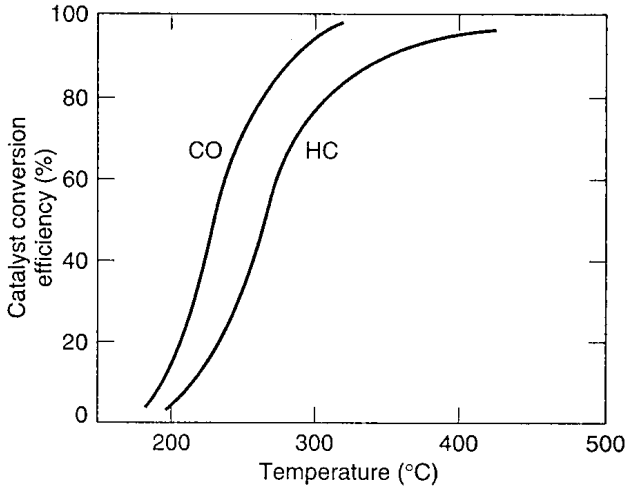

Figure 3.9: Conversion efficiencies for typical oxidising catalysts [6]

The catalytic conversion efficiency is the ratio of the rate of mass removal in the catalyst of the particular constituent of interest to the mass flow of that constituent into the catalyst: eg, for HC.

$$\eta_{CAT} = \frac{\left(\dot{m}_{HC,in} - \dot{m}_{HC,out}\right)}{\dot{m}_{HC,in}} = 1 - \left(\frac{\dot{m}_{HC,out}}{\dot{m}_{HC,in}}\right) \qquad [3.3.1]$$

Or can be expressed in concentration terms as:

$$\eta_{CAT} = \frac{\left([HC]_{in} - [HC]_{out}\right)}{[HC]_{in}} = 1 - \left(\frac{[HC]_{out}}{[HC]_{in}}\right) \qquad [3.3.2]$$

where [ ] in equation 3.3.2 denotes specific emission concentrations in ppm/kW.

The temperature at which a catalytic converter becomes 50% efficient is defined as the light-off temperature. The light-off temperature is about 270^{o}C for the oxidation of HC and 220^{o}C for the oxidation of CO. The conversion efficiency at fully warm conditions is about 98% to 99% for CO and 95% for HC, depending on the HC components. Various measures have been tried to decrease the converter warm up time, including use of an afterburner, locating the converter or an additional start-up converter closer to the exhaust manifold and electric heating.[6]

With diesel engines, catalytic converters are used to oxidise the HC and CO, but reduction of the exhaust nitric oxides is poor because the engine runs lean. Thus, this pollutant has to be controlled by design of the combustion process and/or the choice of operating conditions. The figure below shows experimental results for a direct injection diesel engine.[6]

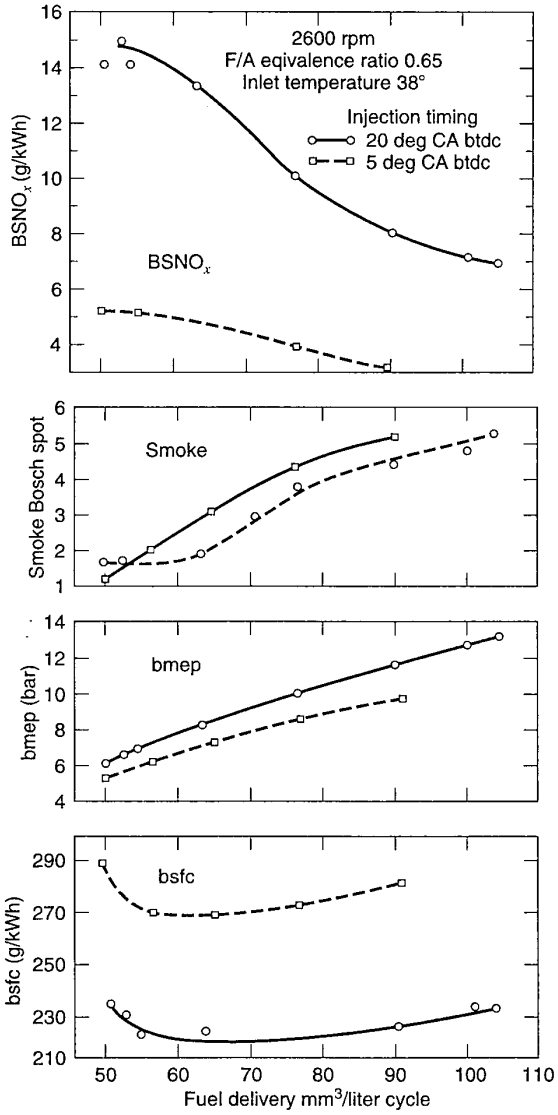

Figure 3.10: Effect of timing on emissions and performance, supercharged research diesel engine, at constant fuel-air ratio and constant air inlet temperature [6]

Notice in figure 3.10, as the injection timing is retarded from 20^{o} to 5^{o} before top dead centre, the nitric oxides drop by about a factor of 3, whereas the fuel consumption increases only about 15%. For this reason, diesel engines are usually operated at injection timings slightly retarded from that which produces best fuel economy.[6]

3.4 Fuel Injection Characteristics

3.4.1 Fuel-Air Distribution in the Spray

A study of the photographic films of Scott[9] showed that at the beginning of combustion the spray has a shape similar to that schematically drawn in figure 3.11 below.

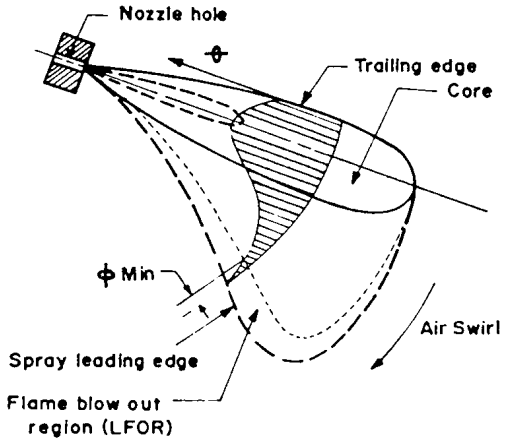

Figure 3.11: Schematic diagram for a fuel spray injected in swirling air [8]

The average distance between the droplets is expected to change with the location in the spray, and is greatest near the leading edge (downwind), where the smaller droplets are concentrated. The average local fuel-air ratio and consequently the combustion mechanisms are therefore expected to vary from one location to the other. Local equivalence ratio and taking into consideration all the fuel present – whether in liquid or vapour phase - produces the distribution shown in figure 3.11 above. This distribution varies with the radial distance from the nozzle hole. However, at the leading edge (downwind) of the spray and for all radii, the fuel-air ratio always approaches zero and increases when approaching the core of the spray. It can also be assumed that most of the droplets, which are carried away from the core will be completely evaporated before the start of ignition[10] or that the mixture near the leading edge of the spray consists of pre-mixed fuel vapour and air.[8]

3.4.2 Fuel deposited on the walls

Some fuel sprays impinge on the walls. This is especially the case in small, high-speed, direct injection engines because of the shorter spray path and the limited number of sprays. The rate of evaporation of the liquid film depends on many factors including gas and wall temperatures, gas velocity, gas pressure and properties of the fuel. Previous studies by El Wakil[11] showed that the liquid droplets reach an equilibrium temperature during evaporation and remain at this temperature until they are completely evaporated. For the liquid film, the local wall temperature may be lower than the equilibrium temperature corresponding to the gas pressure. Therefore, the rate of evaporation of the liquid film is expected to be less than the corresponding rate of droplets, and it is expected that this liquid film will be the last to be evaporated. The vapour concentration is at a maximum on the liquid surface and decreases with increased distance from the surface. An assumption that, if the surrounding gas has a high relative velocity and contains enough oxygen, the flame will propagate to within a small distance from the wall. Combustion of the rest of the fuel on the walls will depend upon the rate of evaporation and mixing of fuel and oxygen. If the surrounding gas has a low oxygen concentration or the mixing is not appropriate, evaporation will occur without complete combustion. Under this condition, the fuel vapour will decompose and form unburned hydrocarbons, partial oxidation products, and carbon particles.[8]

As the piston moves on the expansion stroke, the gases flow outward radially in an inversed squish motion to fill the expanding space between the piston top and cylinder head. In shallow-bowl combustion chambers, most of the combustion process takes place in the bowl. In deep-bowl types, the reversed squish flow is significant and will help the mixing of incomplete combustion products and air during the expansion stroke. In both cases the swirl motion will continue but, due to frictional losses, at a lower rotational velocity than that on the compression stroke. The combination of the inversed squish and swirl produce heterogeneous eddies and tends to draw the fuel vapour and the partial oxidation products, including carbon which has been formed from fuel deposited on the bowl walls. Photographic films by Scott[9], show eddies and heavy smoke clouds outside the bowl from the areas where the spray impinges on the wall. These eddies are heterogeneous, and upon combustion they burn with a luminous flame due to the presence of carbon particles. The progression of combustion in the spray and the pressure and temperature development in the cylinder may be represented as shown in figure 3.12.

The emission formation expected in the different regions of the spray without fuel impingement on the walls may be summarised as shown schematically in figure 3.13.[8]

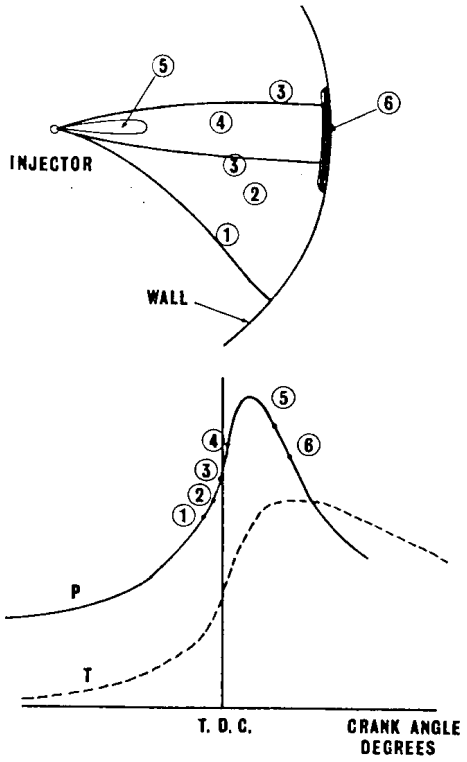

Figure 3.12: Progression of spray combustion, and cylinder pressure and temperature [8]

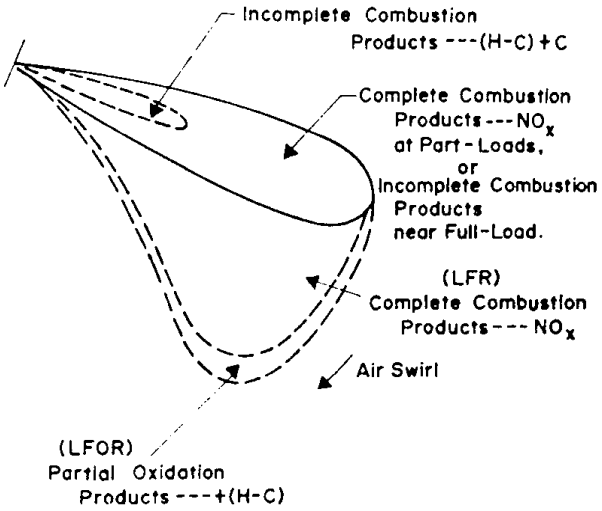

Figure 3.13: Emission formation in a fuel spray injected in swirling air [8]

3.5 Fuel Injection in Diesel Engines

The theme of this paper; is Glikins' [12] illustration in his words, that the fuel injection equipment is the heart of the diesel engine. He explains the task of the fuel injection equipment (FIE), present-day systems and sets out the problems in optimising the injection characteristics.[12]

3.5.1 Introduction

The theme is that the fuel injection equipment is the heart of the diesel engine, not only in the obvious sense that if the fuel injection equipment stops, so does the engine, but that most engine characteristics are directly dependent on the fuel injection system. The engine torque curve, fuel consumption, smoke, noise and emissions are determined by the quantity and way in which the fuel is injected into the engine combustion chamber. In fact, development of the diesel engine itself has been linked very closely to what it has been possible to achieve with the fuel injection equipment.

3.5.2 The Task of the Fuel Injection Equipment

In a diesel engine the fuel is not pre-mixed with the air as in the petrol engine, but is injected directly into the combustion chamber near top dead centre, and ignition occurs spontaneously. The FIE therefore has to fulfil three functions:

1. to meter the quantity of fuel to be injected;
2. to arrange that the injection occurs at the right time in the cycle. This implies accurate timing of injection as a function of engine speed and load;
3. to mix the fuel with the air in the short period of time available. This means that the fuel has to be introduced at very high energy levels.

The quantity of fuel injected into the engine cylinder determines the torque that is generated and therefore accurate control of the quantity injected is necessary, both from cylinder to cylinder, and over time.

An accuracy of timing of the order of +/-1^{o} crankshaft is required to optimise the engine performance, smoke, noise and emissions. Finally, to mix the fuel with the air, high injection pressures are required.

The mixing of the fuel and air immediately after injection is one of the primary factors controlling combustion. This mixing can be achieved by having either a high swirl, as in indirect injection engines, and a maximum injection pressure of about 300 - 400 bar, or low swirl as in direct injection engines and maximum injection pressures of 450 – 850 bar. On quiescent direct injection engines, maximum injection pressures over 1000 bar are required.

3.5.3 The Relationship between Fuel injection and Engine performance

The fuel injection system has an important influence on the following engine parameters:

1. power output
2. specific fuel consumption
3. smoke
4. noise
5. oxides of nitrogen (NO_x)
6. unburned hydrocarbons (HC)
7. particulates

The first four parameters have always been of importance in diesel engines. NO_x and HC are currently under legislation in the USA, Japan and Europe. Legislation of particulates is also currently being implemented in these countries. The discussion that follows is related to swirl DI engines, but many of the conclusions are of wider application. The heat release diagram of a typical diesel engine is shown overleaf.

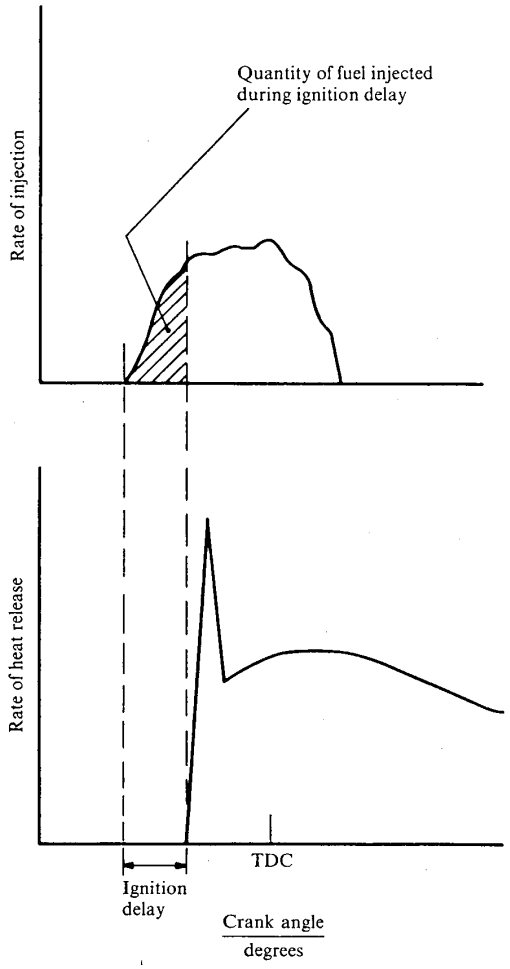

Figure 3.14: Rate of injection and rate of heat release [12]

A factor that is of critical importance in combustion is ignition delay, this is the time between the beginning of ignition and the beginning of combustion. Its magnitude depends on the time taken to achieve a small amount of stoichiometric fuel/air mixture, and the time for pre-flame reactions, which lead to ignition. The length of the ignition delay is dependent on the instantaneous cylinder compression temperature at the point of ignition: the higher the temperature, the shorter the ignition delay. The length of the ignition delay affects the pattern of burning. The fuel injected during the delay period (shown shaded in the above figure) mixes with the air during the ignition delay period and forms a pre-mixed flame giving rise to a rapid heat release and rise in cylinder pressure. The fuel injected after the ignition delay burns as a diffusion flame and gives a more gradual rate of heat release.

The specific fuel consumption and combustion noise are related to the heat release diagram. The formation of smoke and NO_x depend on the local rates of fuel/air mixing as well as the heat release diagram. Specific fuel consumption, smoke, noise and NO_x will now be considered in more detail.

Specific Fuel Consumption

The specific fuel consumption (sfc) is a direct function of the rate of heat release, and would theoretically be a minimum if all the heat were released instantaneously at top dead centre (TDC). In practice, sfc is reduced as the rate of injection (and hence rate of heat release) is increased, but tests have shown[13] that there is a maximum injection rate beyond which no further reduction in sfc is obtained. This rate is known as the maximum useful rate (MUR) and on any particular engine depends on the swirl ratio (higher swirl gives lower value of MUR). Figure 3.15 below shows the sfc as a function of timing as the injection rate is increased.

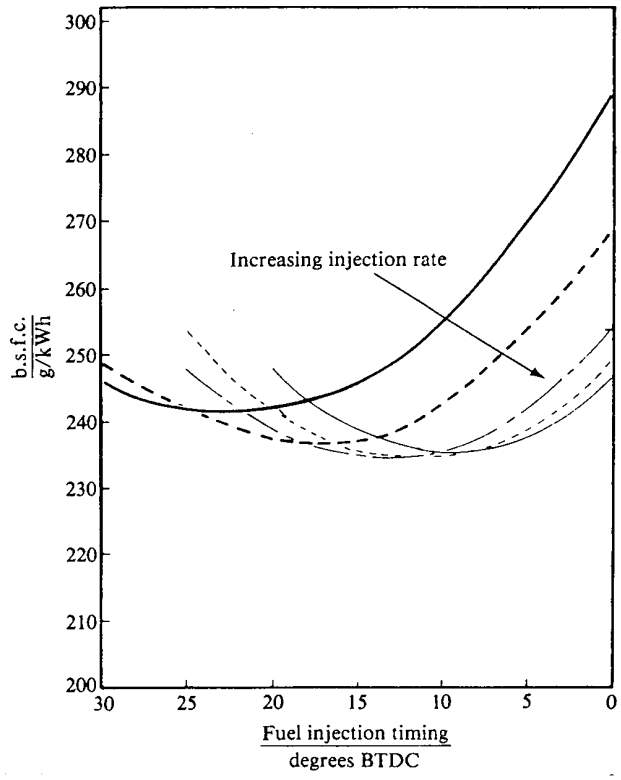

Figure 3.15: Specific fuel consumption as a function of timing and rate of injection [12]

This shows that as the rate is increased, the timing for minimum sfc becomes progressively later.

Smoke

Black smoke arises from the diffusion phase of the combustion. The premixed combustion does not generate black smoke. During the diffusion burning, smoke generation is increased if high temperatures are reached in fuel-rich zones of the spray, and this is most likely to occur shortly after the ignition delay[14,15]. Therefore, improved air/fuel mixing (i.e. higher rate of injection or higher swirl) reduces smoke, as does turbocharging particularly when combined with intercooling.

Noise

Combustion noise is principally a function of the rate of rise of cylinder pressure and this in turn is related to the magnitude of the first peak in the heat release diagram[16]. Plotting maximum rate of heat release against combustion noise gives figure 3.16.

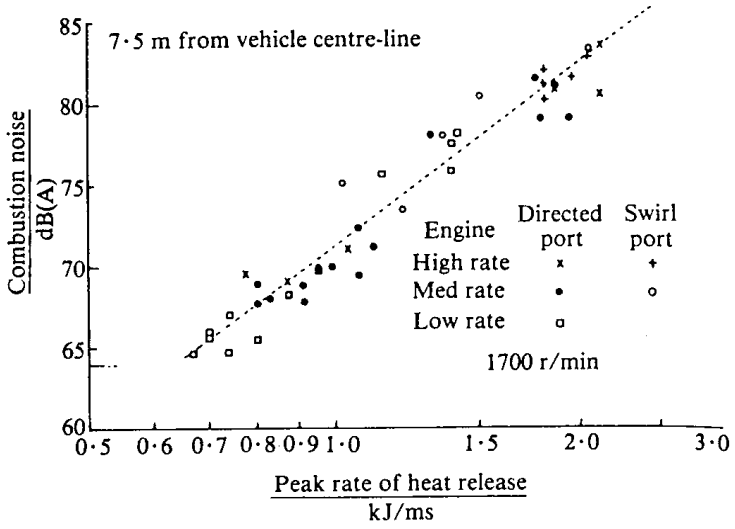

Figure 3.16: Relationship between combustion noise and peak rate of heat release [12]

Since the peak in the heat release diagram depends on the fuel injected during the delay period, it follows that the way to reduce combustion noise is by reducing the ignition delay and controlling the rate of injection during the delay period.

It should be noted that the total engine noise is the sum of the combustion and the mechanical noise from the engine.

Oxides of Nitrogen (NO_x)

Nitrogen oxides arise from the oxidation of atmospheric nitrogen. The amount of NO_x formed increase if high temperatures exist in fuel-lean zones of the spray. In this respect, the formation of NO_x is the inverse of smoke, which, as mentioned before, is formed when high temperatures exist in fuel-rich zones of the spray. It is therefore not surprising that factors that reduce smoke (eg. high injection rate) have the opposite effect on NO_x. However, it may be possible, by increasing the injection rate and retarding the timing, to have better smoke and better NO_x, as seen from the trade off curve in figure 3.17 below.

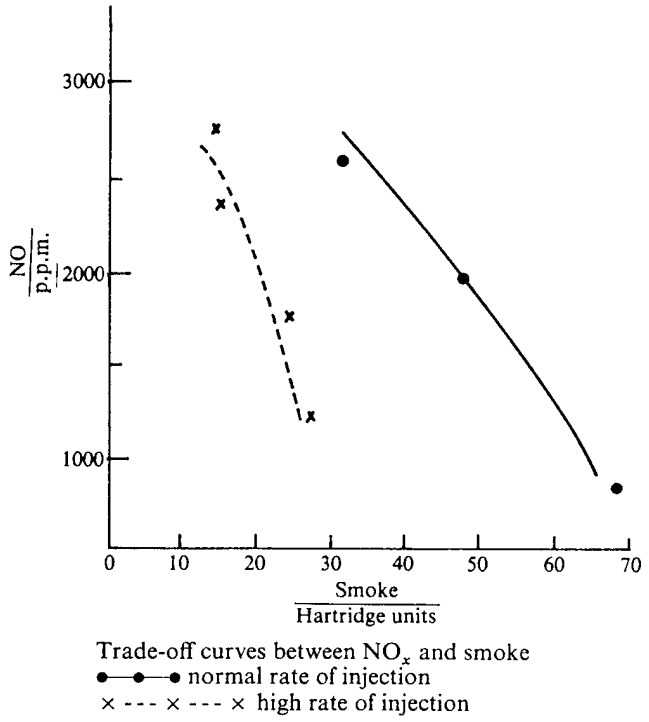

Figure 3.17: Trade off curves between NO_x and smoke [12]

It should be noted that since NO_x is also formed at part load when the smoke is low, it is possible to reduce the total NO_x formed during a driving cycle by using exhaust gas recirculation, without affecting the maximum smoke level.

Since sfc, smoke, noise and NO_x do not change in the same direction when changes are made in rate of injection, timing etc., optimisation of the engine performance is not easy. One way of considering these parameters simultaneously is to plot a set of trade off curves[17] as illustrated below.

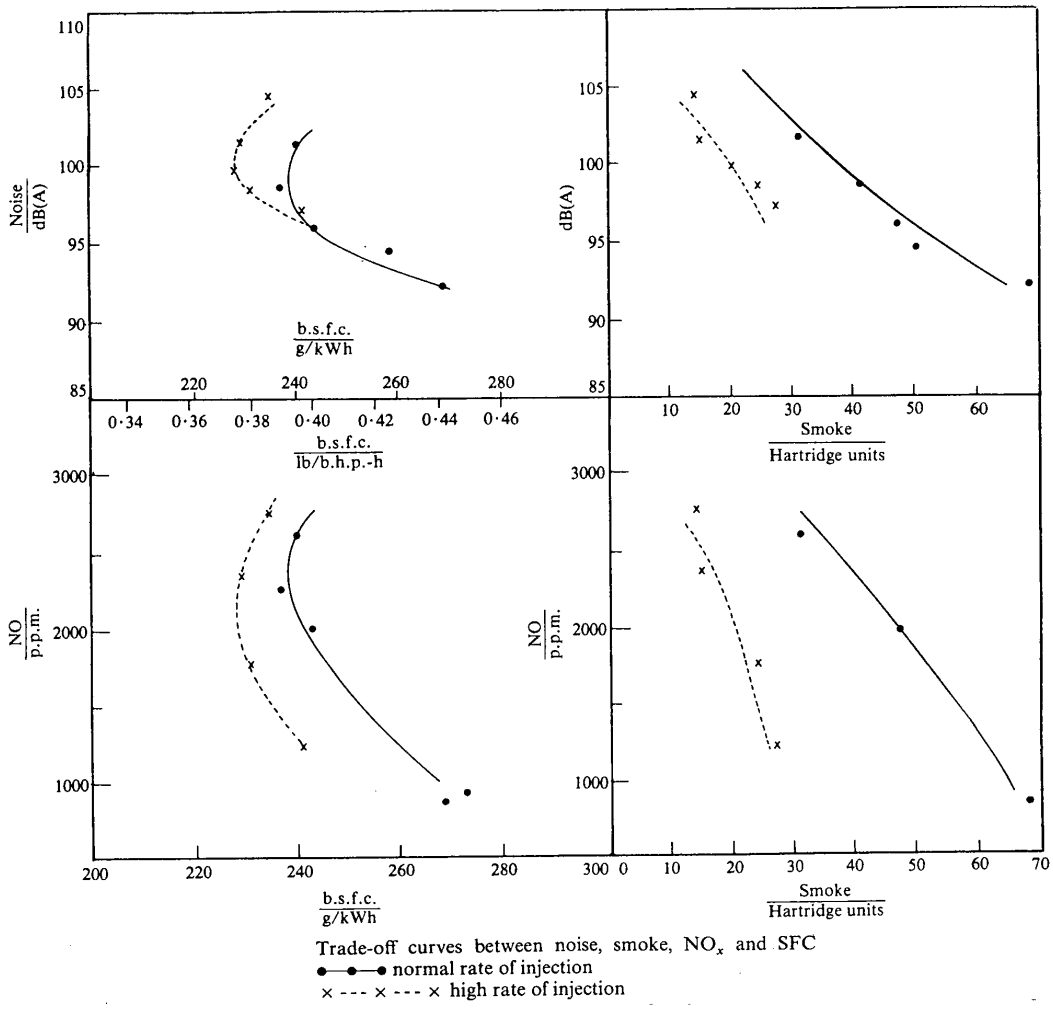

Figure 3.18: Trade off curves between noise, smoke, NO_x and sfc [12]

These show the effect of two rates of injection at different timings on sfc, smoke, noise and NO_x. It can be seen that, for this engine condition, the higher rate of injection can with suitable timing, give the same noise as the lower rate of injection, and improved sfc and NO_x.

Unburned Hydrocarbons (HC)

The origins of HC, have been studied by various researchers and has been shown [15,18] that the formation of HC in DI engines arises from three sources:

1. Too-lean-to-burn mixtures – These can occur during the ignition delay period. As fuel is injected during the ignition delay period, more and more air mixes with the spray and the mixture strength in the periphery of the spray becomes weaker. The longer the delay period, therefore, the greater the amount of 'lean-limit' hydrocarbons. It has been found by experiment that as the delay period is reduced, there is a critical value below which there is no further reduction of HC emission and hence below which the contribution from the lean limit sources becomes negligible.

2. Nozzle sac volume source – After the end of the main injection of fuel when the nozzle needle has seated, there is a small quantity of fuel trapped in the nozzle tip. Some of this fuel escapes into the combustion chamber late in the engine cycle and forms HC emissions. It has been found that HC emission from this source correlates with the volume in the nozzle spray holes.

3. Fuel-rich mixtures late in the engine cycle – Experiments have shown that when the fuel delivery is increased HC remains relatively constant until a certain value of fuel delivery is reached, afterwhich HC increases very steeply with further increase in fuel delivery. It is assumed that this is due to the formation of fuel-rich mixtures late in the engine cycle. For well-optimised engines, this limit is reached above maximum fuelling and HC emission from this cause should not normally take place.

In summary, therefore, to reduce the HC emission, a short ignition delay should be aimed for, and the volume of fuel below the nozzle seat should be minimised.

Particulates

Particulates are made up of a solid phase of soot particles and a liquid phase of unburned hydrocarbons that condense in the engine exhaust pipe below 52^{o}C.

In one investigation[15] on an IDI and a DI engine, it was found that, over a range of engine loads and speeds at different timings, a good correlation was obtained between particulates on the one hand, and smoke and HC on the other.

The following relationship was found:

particulates = 1.0 x smoke + 0.5 x HC measured units in g/m^3

This relationship may not hold over a wider variety of engines or fuel types. However, it seems likely that particulates can be reduced by measures that reduce smoke and HC.

3.5.4 The Requirements imposed on FIE

Having considered the various engine parameters separately, the requirement is to see if it is possible to derive a list of ideal FIE requirements. It is not possible to present such a requirement in a strictly quantitative manner, since the differences between individual engines sometimes require substantially different FIE characteristics, even though the engines may be of similar size and have a similar construction. Nevertheless, certain general ideals for combustion can be specified, and typical characteristics defined. These apply to IDI as well as DI engines.

There are three ideals for optimising combustion[15]. These are:

1. To achieve ignition with a minimum amount of premixed fuel. This is to reduce the lean limit HC, the liquid phase of particulates, and to reduce noise.
2. To mix the fuel after ignition at as high a rate as permissible. This will minimise the burn time and hence reduce sfc. The higher mixing rate in the diffusion phase of combustion will reduce smoke. However, there is a maximum permissible rate of fuel/air mixing because as the rate of mixing increases, so does the cylinder temperature and pressure, thus increasing NO_x and noise.
3. To optimise the timing of combustion with respect to crank angle across the speed and load range of the engine.

These requirements, particularly point (3), together with various refinements to improve the overall acceptability of diesel engines, suggest a need for a more accurate and flexible control of injection than may be possible through conventional hydro-mechanical means alone.[12]

3.6 Improvement in Performance of a Methanol Fuelled Compression Ignition Engine

The University of the Witwatersrand has been performing extensive research in the field of combustion of alcohol fuels. Methanol is one of the fuels under investigation and this paper[19] illustrates the results of the research conducted. Ethanol and methanol are both alcohols, thus various aspects present in methanol research results would to a certain degree also be prevalent in ethanol.

3.6.1 Introduction

The performance and emissions of aspirated dimethyl ether as ignition promoter, and methanol as main fuel in a compression ignition engine were investigated. The ignition quality of methanol is very poor hence is assisted by dimethyl ether. Previous work has been done by the University to investigate the combustion, performance and emissions of methanol with dimethyl ether as ignition promoter [20,21,22]. The compression ignition engine used consisted of a standard two-cylinder, water-cooled, direct injection engine. The emissions species measured included NO_x, THC, CO and CO_2. The tests were first performed on diesel fuel at a constant speed, while the load was increased. The testing procedure was then performed at the same speed with methanol/DME fuel. The engine setting namely the injector pressure and injection timing were set for optimum performance on diesel fuel, and remained unaltered for methanol/DME. The reason for this was to study the effect of switching fuels without alterations, thus simplifying the changeover from diesel to methanol.

3.6.2 Results and Discussion

The results are discussed under two aspects, namely performance and emissions. Constant speed tests were performed at 1550 rpm on diesel fuel. The load was increased from zero in small increments until a smoke level of over 75 HSU was reached. When the engine was fuelled on methanol, the load was also increased in the same increments as diesel, however was stopped when the engine was no longer able to maintain the selected speed. The amount of total fuel was increased on each test, but the percentage DME to total combustibles was decreased as the load increased.

The proportion of DME to total combustibles was slightly higher than in previous tests. The engine was found to run more smoothly, and fewer speed fluctuations were experienced.

3.6.2.1 Performance

The performance of the engine was considered in terms of brake power, brake specific fuel conversion efficiency, maximum combustion chamber pressure and maximum combustion chamber temperature. Figure 3.19 shows the brake power for both fuelling systems, against equivalence ratio ϕ.

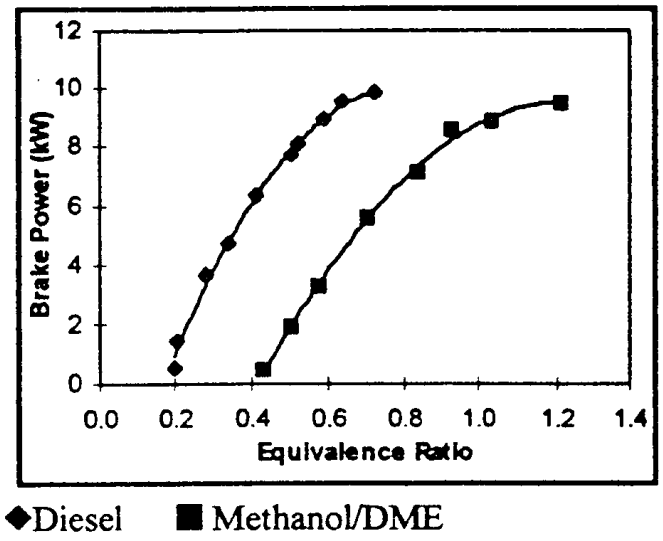

Figure 3.19: Brake power vs. Equivalence ratio [19]

The equivalence ratio for diesel fuelling extends from about 0.20 to 0.72, while that of methanol/DME ranges from 0.43 to some 1.20. The maximum brake power attained on diesel fuel was about 9.8 kW while that with methanol/DME was less at approximately 9.5 kW. However, this power was reached with methanol/DME at an equivalence ratio of about 1.2. This means that a greater amount of fuel was needed. This is to be expected since the calorific value of both methanol and DME are about half that of diesel. Both curves show an increasing trend with maximum power being reached at the maximum equivalence ratio with both fuelling methods. If more fuel had been introduced, the power could be expected to fall off. This is undesirable, especially in the case of diesel, it would have resulted in an excessive amount of smoke being emitted. This result is not conducive to a clean environment and compliant to emissions standards.

When it is considered that the engine was set for optimum performance on diesel, and the same power was reached with methanol/DME fuelling, it shows that this fuelling combination is a viable replacement for diesel fuel. From the point of view of performance, methanol/DME is advantageous, but the cost of this fuel may at present, outweigh this advantage. Figure 3.20 shows the graph of brake specific fuel conversion efficiency versus equivalence ratio for both fuelling methods.

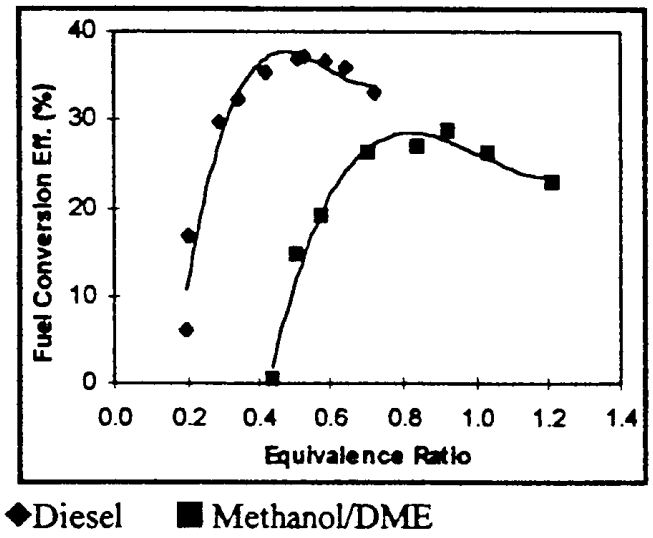

Figure 3.20: Brake Specific fuel conversion efficiency vs. Equivalence ratio [19]

Both graphs show an increasing trend with increasing ϕ, until a maximum is reached, and then a slight decrease is noted. However, the brake specific fuel conversion efficiency with diesel fuelling is better than that with methanol/DME fuelling. The maximum value is some 37% at an equivalence ratio of 0.52 for diesel fuel, and 29% at about 0.92 for methanol/DME fuelling. However, as noted above, the engine was set for optimum performance on diesel fuel. This may well not be the optimum setting for methanol. A higher efficiency may be achieved if the engine was optimised for methanol, in terms of injection timing and injection opening pressure.

The maximum combustion chamber pressure versus equivalence ratio is shown in figure 3.21 below, where a similar pattern is noted for both fuelling systems.

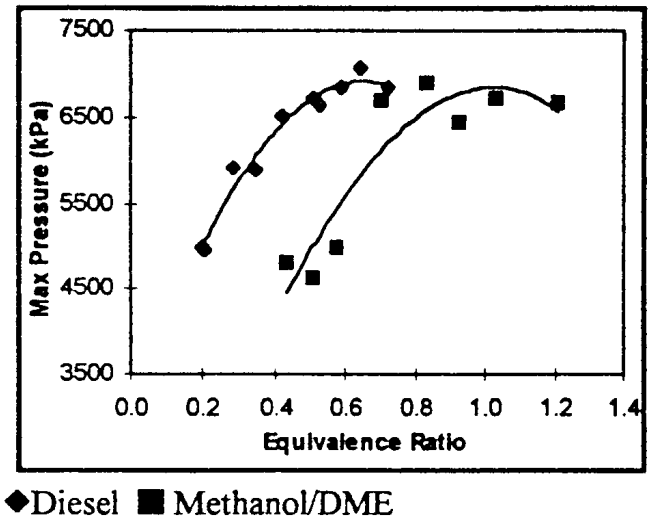

Figure 3.21: Maximum combustion chamber pressure vs. Equivalence ratio [19]

In each instance the pressure rises with increasing load until a maximum is reached. As the amount of fuel is increased further, the combustion chamber pressure begins to decrease. The maximum pressure achieved with both fuelling methods was some 70 bar. The same pressure was not achieved in previous tests. This is believed to be due to the simpler method introduced of gravity feeding the fuel, where there is no possibility of the fuel return line from the injectors being pressurised by the additional pump, which had previously been installed.

The maximum combustion chamber temperature versus ϕ, shown in figure 3.22, displays the same trend as that of the pressure shown in figure 3.21.

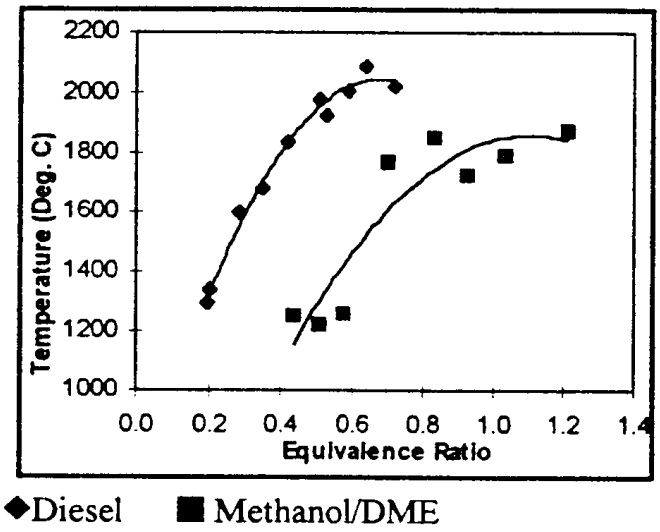

Figure 3.22: Maximum combustion chamber temperature vs. Equivalence ratio [19]

The maximum temperatures reached were just under 2100 ^{o}C for diesel fuelling and about 1900 ^{o}C for methanol/DME fuelling. However, in the case of diesel the maximum temperature coincides with the maximum combustion chamber pressure in terms of equivalence ratio. Instead, the maximum temperature, with methanol/DME fuelling, is reached in the last test. The maximum temperature occurs later than the maximum pressure, as a result of the energy release pattern of the DME. It has been shown previously [20,21,22] that DME releases energy in two stages. The first stage releases an approximately constant amount, while the second stage releases an increasing amount with increasing equivalence ratio. The second stage thus releases the balance of the energy in the DME. Since the amount of DME increases with ϕ, the second stage of energy release also increases in magnitude. Thus the combustion chamber temperature can also be expected to increase with equivalence ratio.

3.6.2.2 Emissions

Attention is now focused on some emission species, namely CO, NO_x and total hydrocarbons, THC.

Figure 3.23, shows CO emissions for both fuelling systems versus equivalence ratio.

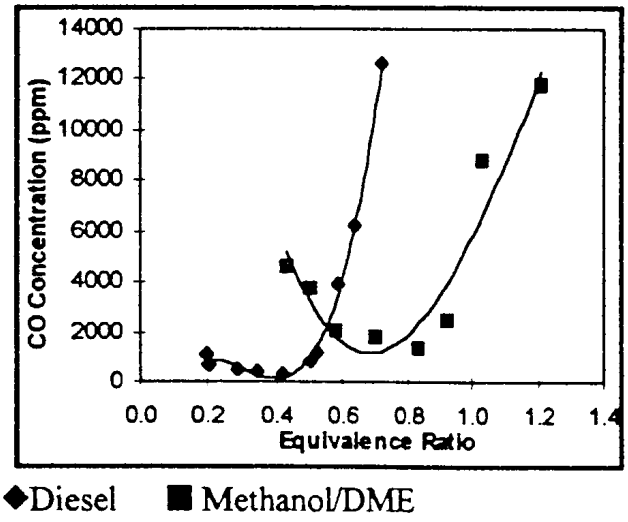

Figure 3.23: CO concentration vs. Equivalence ratio [19]

The pattern exhibited by the diesel graph is as expected, in that the concentration at no load is low, at about 1100 ppm. It then decreases further, reaching a minimum of about 380 ppm, before increasing steadily to over 12000 ppm, at the highest load. The concentration of CO for methanol/DME starts somewhat higher, at some 4600 ppm, and decreases to a minimum of 1400 ppm at a ϕ value of 0.835. The CO concentration then increases steadily to just under 12000 ppm at a ϕ value of 1.2. It is noteworthy that where the CO concentration for diesel fuel is at its maximum, that of methanol is at its minimum. This means that if the engine was to be fuelled on methanol/DME and run at a ϕ value of about 0.80 to 0.85 it would represent a considerable advantage in terms of CO emissions. Furthermore, the point of minimum CO concentration is very close to the point of maximum brake specific fuel conversion efficiency.

The next emission specie to be discussed is that of NO_x as shown in figure 3.24. The trends shown are almost identical, reaching a maximum and then decreasing.

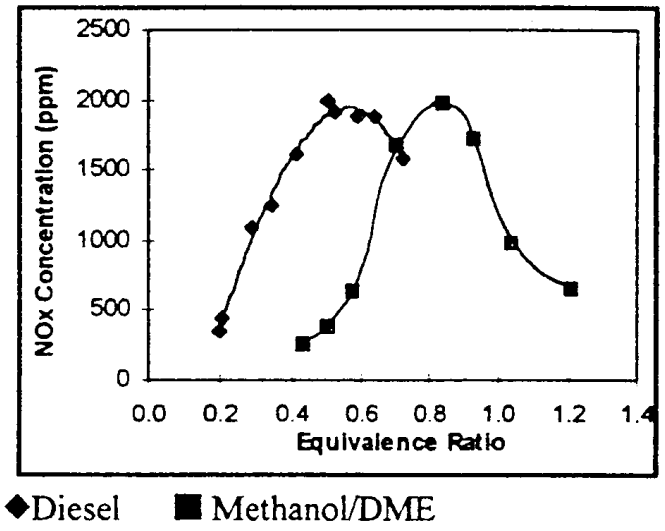

Figure 3.24: NO_x concentration vs. Equivalence ratio [19]

At maximum load the reduction is greater in the case of methanol/DME, where a value of about 650 ppm was reached at an equivalence ratio of 1.2 against some 1600 ppm at 0.72 for diesel fuelling. A similar situation to that of CO is also noted. When the NO_x emissions for diesel fuel are at their maximum, about 2000 ppm, those of methanol/DME are only about 700 ppm. However, at this stage, the brake power of the engine is considerably less in the case of methanol/DME. Having extended the equivalence ratio range in the instance of methanol/DME, the NO_x emissions at an equivalence ratio of about 0.8, have also shown an increase, to the point where the maximum concentration is virtually the same as that of diesel fuelling. However, at low and high loads, the concentrations of NO_x are considerably reduced. Since the formation of NO_x is favoured by higher temperatures [19], this pattern can be expected. It has been shown that the maximum combustion chamber pressure and consequently the combustion chamber temperature with methanol/DME are not that much lower than that of diesel fuelling. Thus the concentration of NO_x can also be expected to be high when the engine is fuelled with methanol/DME.

The last exhaust emission under consideration is that of the total hydrocarbons. This particular exhaust species has not been discussed in any previous work, because, as pointed out above, the analyser for this particular gas was not available. The concentration of THC versus equivalence ratio is shown in figure 3.25.

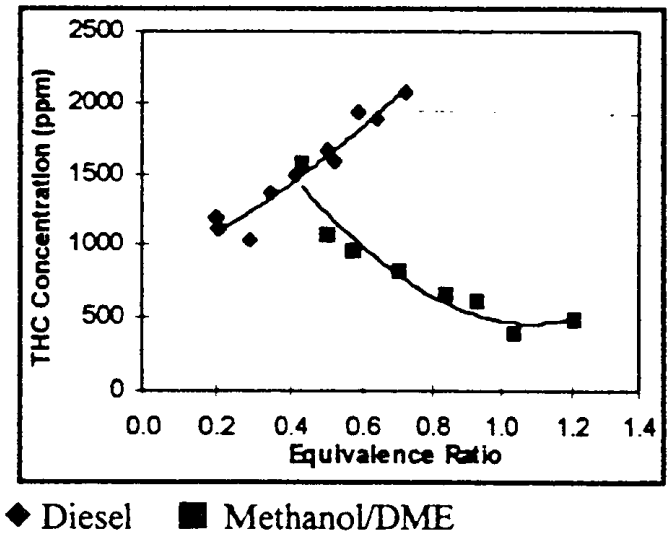

Figure 3.25: THC concentration vs. Equivalence ratio [19]

Two opposing trends are evident. In the case of diesel fuelling, the concentration of THC increases as ϕ increases, whilst in the methanol/DME instance, the THC concentration shows a steady decrease. As before, when the engine was on full load, on diesel fuel, the THC concentration reaches its maximum of almost 2100 ppm, whilst that of methanol/DME is around 800 ppm. However, the greatest benefit is achieved at full load on methanol/DME, where the brake power is virtually the same with both fuelling systems, but the THC emissions are considerably reduced. The concentration of THC decreases from some 2100 ppm at full load on diesel fuel to about 500 ppm on methanol/DME. This represents a considerable benefit in terms of the environment.

3.6.3 Conclusion

The aims of the research conducted were to study the effects of increasing the proportion of DME to total combustibles on the performance and emissions of a compression ignition engine. The results revealed that the brake power with the methanol/DME fuelling almost equalled that developed with diesel fuel. The brake specific fuel conversion efficiency, however, was slightly lower with methanol/DME than with diesel. Considerable improvement was achieved in THC emissions with methanol/DME, where a reduction of some 75% was achieved at maximum load. NO_x emissions showed a reduction of some 60% at maximum load on the methanol/DME combination. CO emissions, however showed a very modest reduction at maximum load.

3.7 Reduction of Emissions in an automotive Direct Injection Diesel engine dual-fuelled with Natural gas by using variable Exhaust Gas Recirculation

3.7.1 Introduction

A promising measure that could overcome the increasing difficulty of meeting diesel emission standards in many countries, is the use of alternative fuels such as compressed natural gas (CNG) in diesel engines. In this method, natural gas is pre-mixed with ambient air in the intake pipe, inducted into the cylinder, and then subjected to multi-point ignition as a result of compression ignition and combustion of pilot diesel fuel sprays. Dual-fuel operation with CNG fuel can significantly reduce smoke. For reduction of NO_x, there are several methods such as retarding injection timing, water injection, multi-stage injection and exhaust gas recirculation (EGR). At lower loads, by increasing the intake charge temperature, EGR could enhance the combustion process, because some of the unburned fuel can be reburned with this method. Simultaneously, NO_x is reduced and smoke is reduced to almost zero at a high fraction of CNG supply. Therefore, in this study[23], the effects of EGR on a dual-fuelled diesel engine are investigated in terms of the performance and emissions characteristics.

3.7.2 Exhaust Gas Recirculation

In general, EGR is a useful method for reducing NO_x formed in the cylinder, owing to the lowered combustion temperature resulting from the increased inert (unreactive) gas in the fuel charge. Thus it was expected that the combination of dual fuelling and EGR could considerably reduce both smoke and NO_x, especially at higher loads. In addition, unburned hydrocarbons contained in the recirculated gas would possibly reburn in the mixture, leading to reduced unburned fuel and improved thermal efficiency at lower loads. In this investigation, to quantify the amount of EGR, the EGR ratio is defined as the mass ratio of the recirculated gas to the ambient air inducted into the cylinder.

3.7.3 Results and Discussion

3.7.3.1 Effect on Brake Specific Fuel Consumption

Figure 3.26 below shows the bsfc change versus engine speed for diesel and diesel-gas engines with different EGR percentages. As indicated, with increasing EGR percentage, bsfc increases. The reason for this order is the decrease of power with increasing EGR percentage.

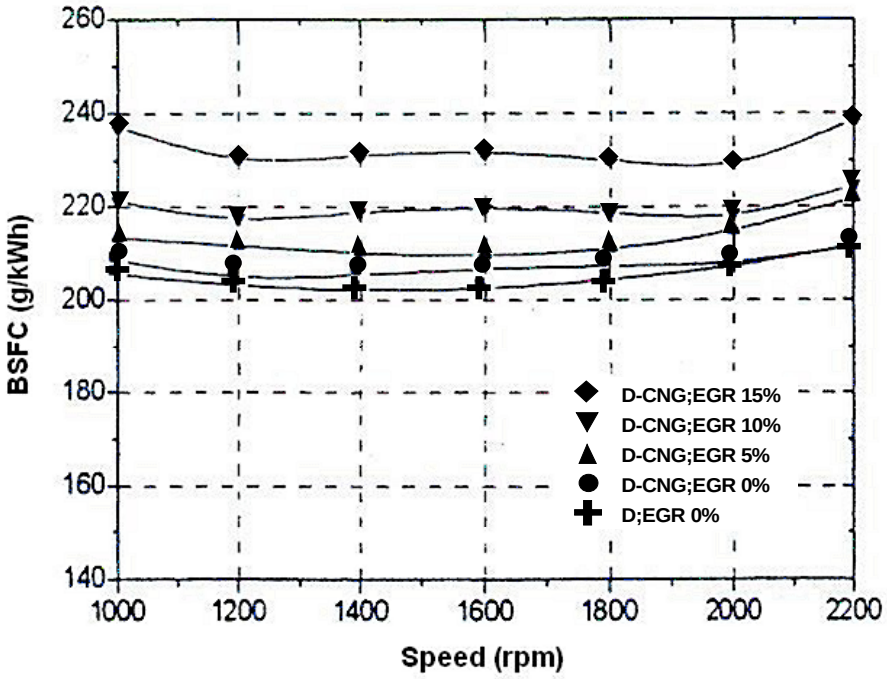

Figure 3.26: Brake specific fuel consumption for various EGR ratios [23]

3.7.3.2 Effect on Unburned Hydrocarbons (UHC) emissions

Figure 3.27 shows the UHC change versus EGR percentages. As indicated, with increasing EGR percentages, UHC is increased, because with increasing EGR percentage, the mixture of fuel and air becomes richer, and so the combustion process relative to zero % EGR deteriorates.

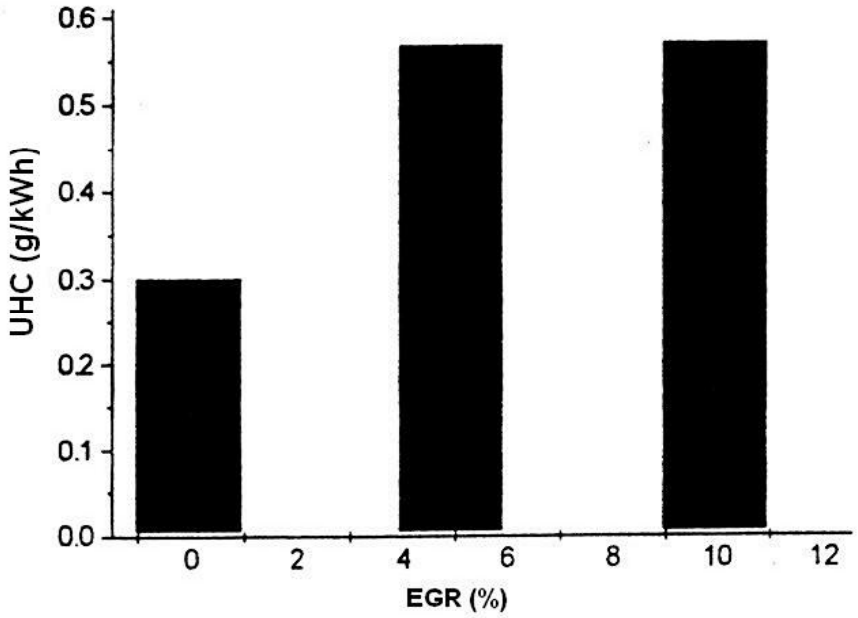

Figure 3.27: Unburned hydrocarbon emissions for increasing EGR ratios [23]

3.7.3.3 Effect on Carbon Monoxide (CO) emissions

Figure 3.28 below shows the CO change versus EGR percentages. As indicated, with increasing EGR percentages, CO is increased, because as in the case of UHC, with increasing EGR percentage, the mixture of fuel and air becomes richer, and the resultant combustion relative to zero % EGR deteriorates.

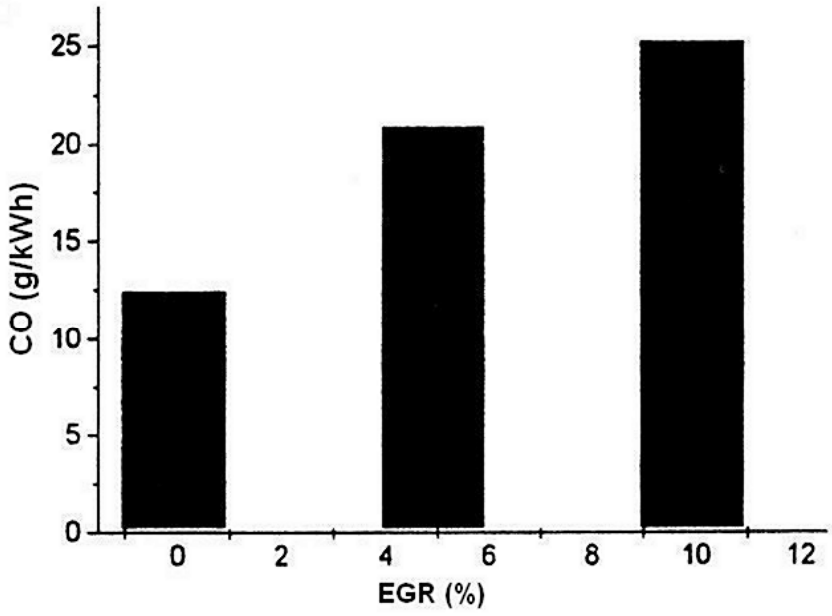

Figure 3.28: Carbon monoxide emissions for increasing EGR ratios [23]

3.7.3.4 Effect on NO_x emissions

Figure 3.29 below shows the NO_x emission change versus EGR percentages. As indicated, with increasing EGR percentages, NO_x is reduced, because EGR reduces the maximum flame temperature and hence NO_x is reduced.

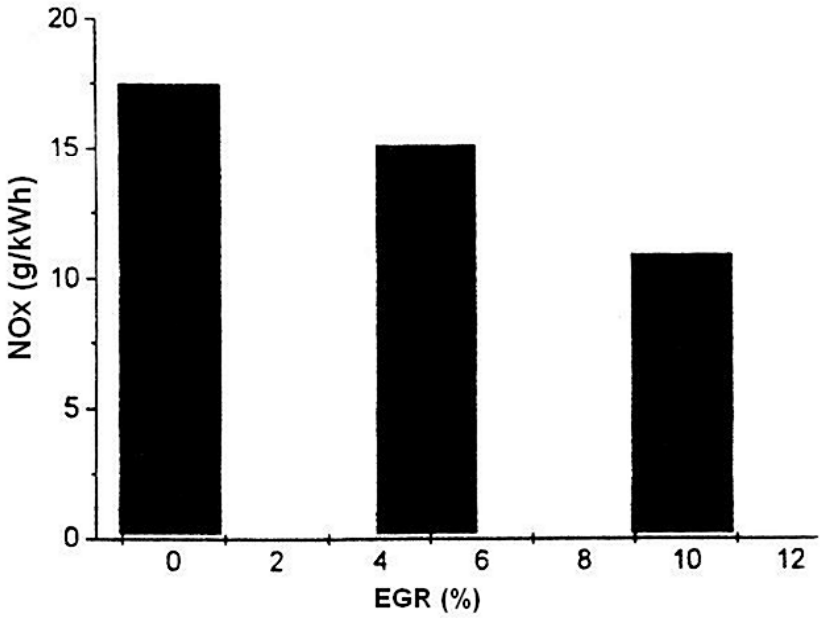

Figure 3.29: Nitrogen oxide emissions for increasing EGR ratios [23]

3.7.3.5 Effect on Soot emissions

Figure 3.30 shows the soot emission change versus EGR percentages. As indicated, with increasing EGR percentages, soot emissions increase. With the use of EGR, the cylinder temperature is reduced. Also, the amount of oxygen in the cylinder is reduced. Both of these parameters increase soot production within the cylinder.

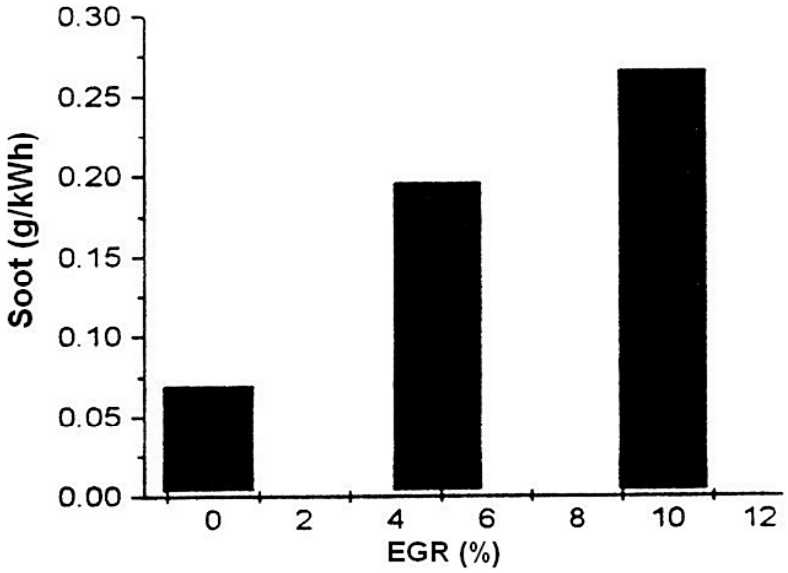

Figure 3.30: Soot emissions for increasing EGR ratios [23]

3.7.4 Conclusion

The results of the investigation revealed that the use of EGR has an undesirable effect on power, bsfc, UHC emissions, CO emissions and soot production, however has a desirable effect on exhaust gas temperature and NO_x emissions.

3.8 Combustion characteristics and Heat release analysis of a Direct injection compression ignition engine fuelled with Diesel-Dimethyl Carbonate blends

3.8.1 Introduction

The reduction in engine emissions and comparative performance parameters to that of diesel is a major research aspect in current engine development. This is in light of increasing concern of environmental protection and preservation. A major prospective method to address this increasing concern, is the use of oxygenated alternative fuels or diesel-oxygenated fuel blends to provide more oxygen during combustion. Some preliminary investigations have been conducted by previous researchers, however, there are still many aspects worth investigating on a quantitative level. These quantitative parameters are expected to supply more information on engine combustion with oxygenated fuels and provide more practical measures for improving the combustion and reduction in emissions. The objectives of this paper are thus to investigate the combustion characteristics of dimethyl carbonate in diesel fuel, and to conduct a systematic analysis of the heat release parameters of a diesel engine operating on diesel-DMC fuel blends.[24]

3.8.2 Fuel Properties

The fuel properties of dimethyl carbonate (DMC) and diesel are given in Table 3.1 below.

Table 3.1: Fuel Properties of Diesel and Dimethyl Carbonate (DMC) [24]

Properties	Diesel	DMC
Chemical Formula	$C_{10.8}H_{18.7}$	$CH_3O(CO)OCH_3$
Mole Weight (g)	148.3	90.1
Density (g/cm^3)	0.86	1.079
Boiling point	180 - 330	90.9
Heat of evaporation (kJ/kg)	270	369
Lower heating value (MJ/kg)	44.8	15.78
Stoichiometric air-fuel ratio	14.4	4.185
Cetane Number	45	35
Carbon content (wt %)	87.4	40
Hydrogen content (wt %)	12.6	6.7
Oxygen content (wt %)	0	53.3

The fuel properties show that DMC has a high oxygen content, while its heat value is lower and its cetane number is slightly lower compared to diesel fuel. In the experiment, oxygenated fuel blends with various proportions of DMC were used on the engine, and combustion characteristics and heat release parameters were analysed at the same bmep. Furthermore, these parameters were compared to those of pure diesel combustion in order to clarify the effect of the oxygenate additive on combustion.

3.8.3 Results and Analysis

Figure 3.31 to 3.34 show the oxygen mass fraction, cetane number, lower heating value and heat of evaporation of the diesel-DMC fuel blends respectively.

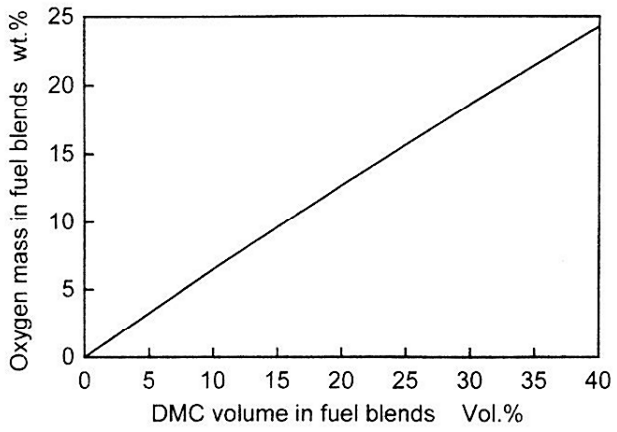

Figure 3.31: Proportion of oxygen in fuel blends [24]

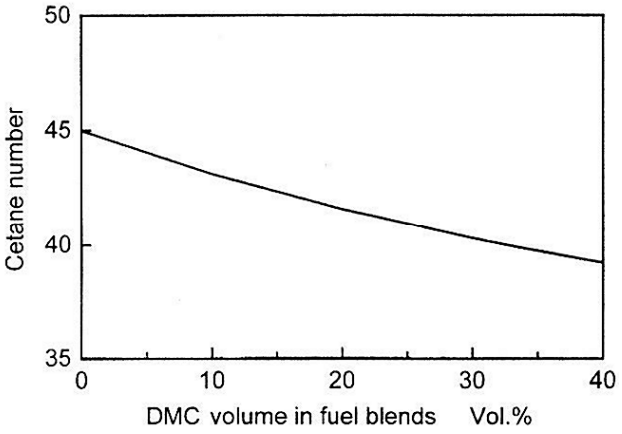

Figure 3.32: Cetane number of fuel blends [24]

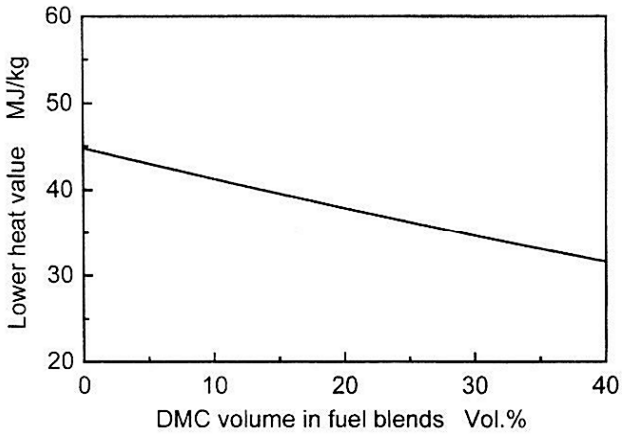

Figure 3.33: Lower heating value of fuel blends [24]

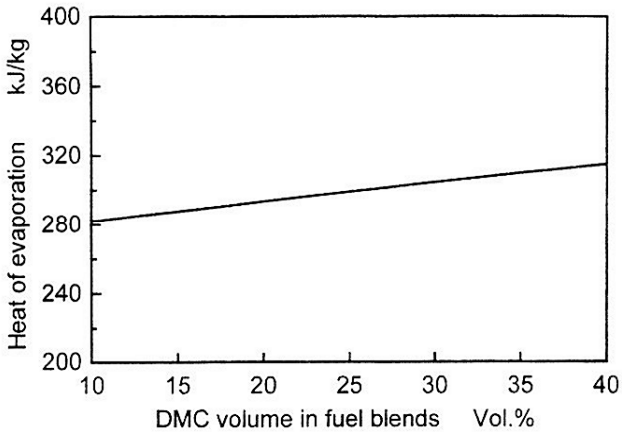

Figure 3.34: Heat of evaporation of fuel blends [24]

The cetane number decreases slightly with increasing DMC addition owing to the lower heating value of DMC, and this will probably lead to an increase in ignition delay of diesel-DMC fuel blends. The lower heating value of the blends decreases and the heat value of evaporation increases with increasing DMC addition, the former factor requiring more fuel to be supplied in order to achieve the same power output, and the latter factor leading to a relatively larger temperature decrease owing to evaporation of the injected fuel.

Figure 3.35 shows the rate of heat release of diesel and diesel-DMC blends at specific bmep.

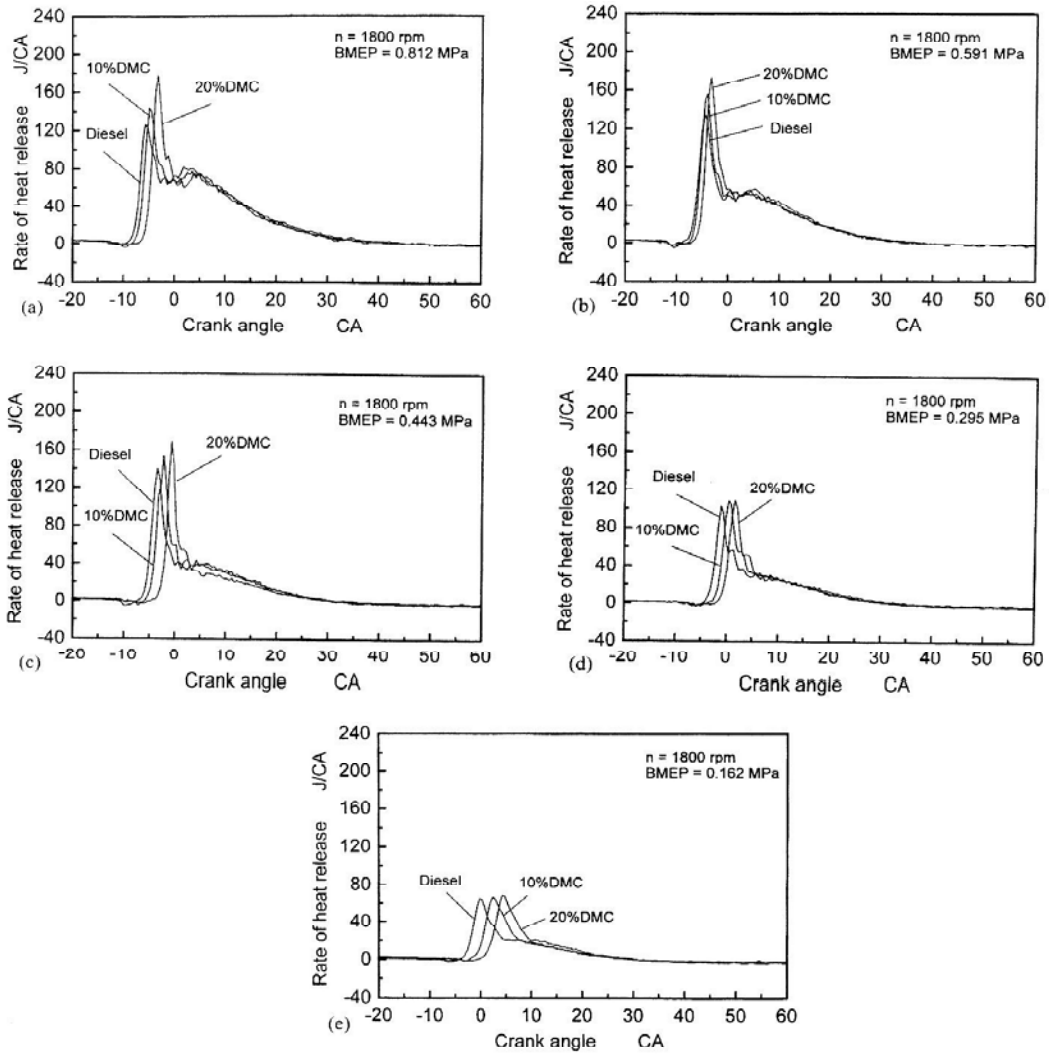

Figure 3.35: Heat release rate characteristics of various DMC additions [24]

The results show that the premixed combustion is prolonged with increasing DMC addition, and the maximum rate of heat release increases with increasing DMC addition. It is suggested that the increase in ignition delay is responsible for this behaviour. Although the beginning of the heat release is retarded for diesel-DMC blends, the end of heat release remains at almost the same crank angle regardless of the DMC volume addition to the diesel fuel. Obviously, the duration of diffusive combustion would decrease with increasing DMC addition, which would be due to the improvement in combustion, as more oxygen would be available. Overall, the mass fraction of fuel consumed in premixed combustion increases and the duration of diffusive combustion decreases with increasing DMC addition.

Figure 3.36 shows the ignition delay, the rapid combustion duration and the total combustion duration for various DMC additions.

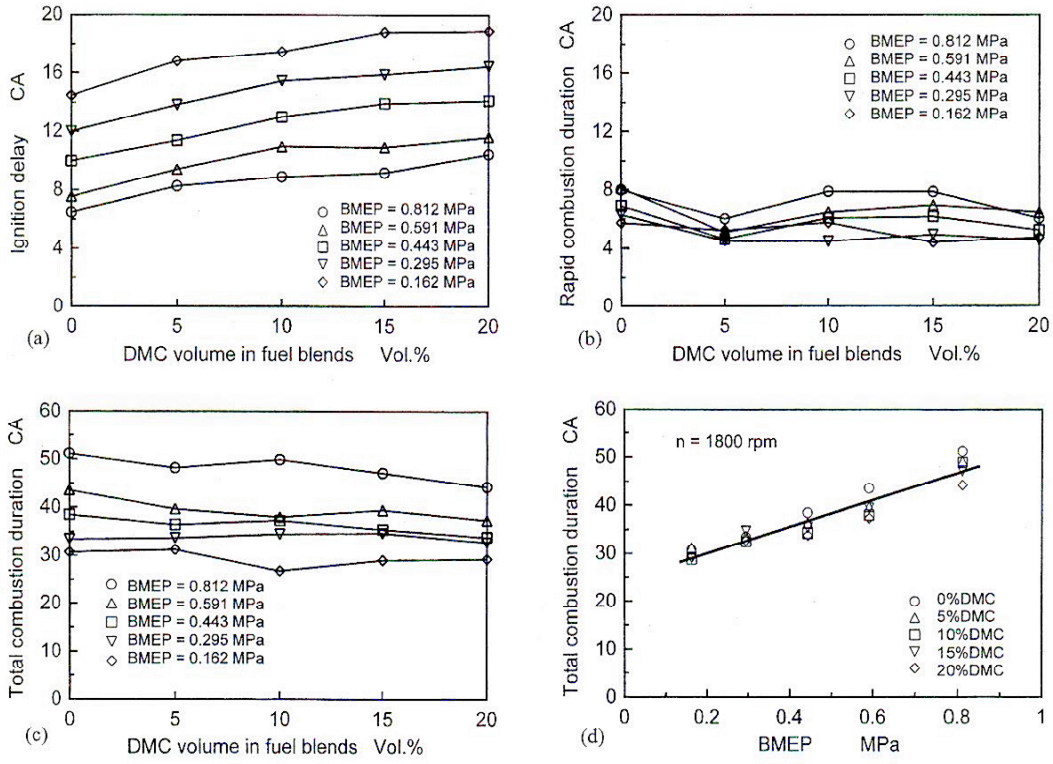

Figure 3.36: Ignition delay, rapid combustion and combustion durations versus DMC additions [24]

The ignition delay is the time interval from the beginning of the nozzle valve lift to the beginning of rapid pressure rise. The rapid combustion duration is the time interval from the beginning of rapid pressure rise to the end of rapid pressure rise. The total combustion duration is the time interval from the beginning of heat release to the end of heat release.

For a specific bmep, the ignition delay increases slightly with increasing DMC addition, while, at a given DMC addition, the ignition delay increases with decreasing load. This is also reasonable since the cetane number of the blended fuels decrease with increasing DMC, and the cylinder gas temperature at the time of fuel injection increases with increasing load. The rapid combustion duration shows slight variations with the addition of DMC, so that, once the combustion is initiated, the rapid combustion period can proceed at the same speed. The total combustion duration shows a negligible change with increasing DMC addition. The combustion duration shows the same linear increasing trend versus bmep for both diesel fuel and diesel-DMC fuel blends.

Figure 3.37 gives the maximum cylinder gas pressure, p_{max} and its corresponding crank angle for different DMC additions.

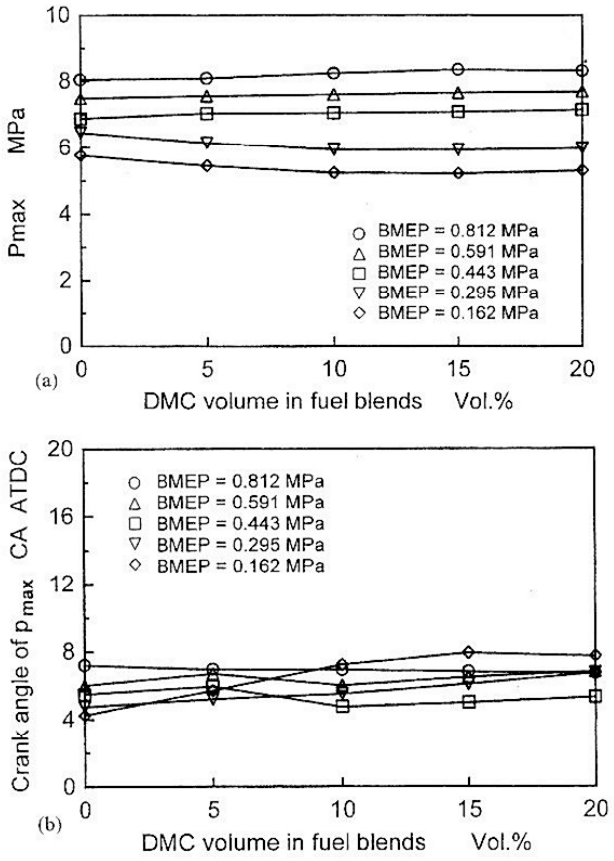

Figure 3.37: Maximum cylinder pressure and its crank angle versus DMC additions [24]

P_{max} shows a slight increase with increasing DMC addition at medium and high loads, whereas it decreases slightly with increasing DMC addition at low loads. The high premixed combustion rate can be attributed to the high value of p_{max} at medium and high loads, while prolonged heat release would be responsible for the decrease in p_{max} at low loads. The crank angle of p_{max} shows a slight increase with increasing DMC addition at low load and less variation with DMC additions at medium and high loads, the high premixed combustion rate at medium and high loads also contributing to this occurrence.

Figure 3.38 gives the bsfc versus bmep for different DMC additions.

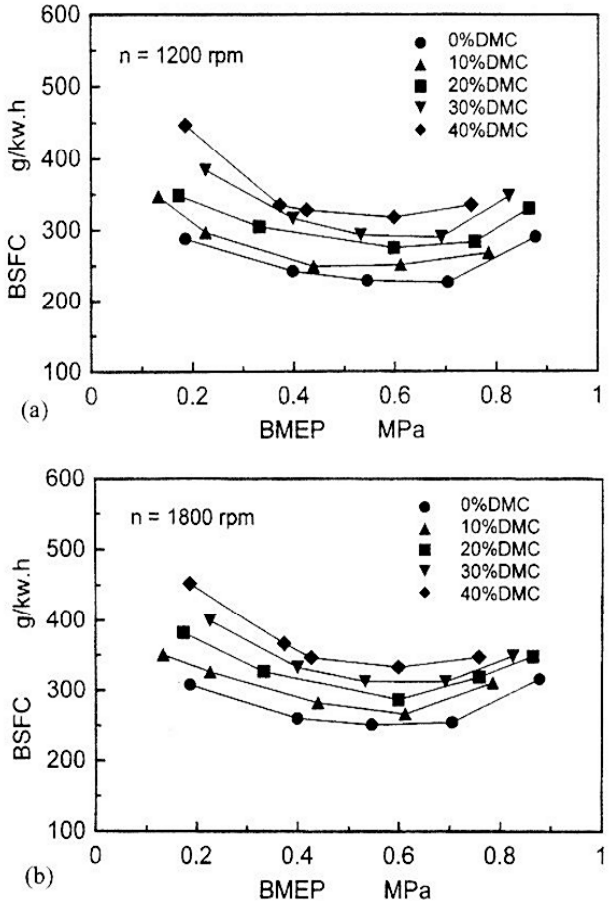

Figure 3.38: Bsfc versus bmep for different DMC additions [24]

For a specific bmep, the bsfc shows an increase with increasing DMC addition, and this behaviour is the same at both 1200 and 1800 rpm speeds. This is reasonable since the engine will consume more fuel with diesel-DMC blends than with diesel fuel to gain the same power output, owing to a decrease in the lower heating value of diesel-DMC blends, and a higher DMC addition causes a larger decrease in the lower heating value, as shown in figure 3.33.

3.8.4 Conclusion

The combustion characteristics and heat release of a DI compression ignition engine fuelled with diesel-dimethyl carbonate blends have been investigated, and the main results can be summarised as follows:

i. Premixed combustion is prolonged and the duration of diffusive combustion is shortened with increasing DMC addition. The maximum cylinder gas pressure increases with increasing DMC addition at medium and high loads, while less variations exist with DMC addition for diesel fuel at low loads.

ii. The ignition delay increases, while the rapid combustion duration and the total combustion duration show less variation with DMC addition.

iii. The bsfc increases with increasing DMC addition.

3.9 Engine performance and emissions of a compression ignition engine operating on Diesel-Methanol blends

3.9.1 Introduction

The reduction in engine emissions is a major research aspect of engine development arising from increasing concern about environmental protection and stringent exhaust gas regulation. With the difficulty to reduce nitrogen oxides (NO_x) and smoke simultaneously in a normal diesel engine, due to the trade off between NO_x and smoke, one prospective method to solve this problem is to use alternative oxygenated fuels or to add the oxygenated fuels to diesel to provide more oxygen during combustion. Practically, adding some oxygenated compounds to fuels seems to be a more attractive option than performing engine modifications, to reduce emissions.

Methanol is regarded as one of the promising alternative fuels or oxygen additives in diesel engines with its advantages of low price and high oxygen fraction. However, owing to the difficulty in forming a stabilised diesel-methanol blend, insufficient research was found on this topic, and previous work mainly concentrated on the application of diesel-ethanol blends in a compression ignition engine. Therefore, much work is needed on the application of diesel-methanol blends in a compression ignition engine to clarify the basic combustion and emissions characteristics and to provide an approach for attaining a stabilised diesel-methanol blend with some solvent. The objectives of this study is thus to form a stabilised diesel-methanol blend by adding a specific solvent and then to investigate the performance and emissions of a compression ignition engine operating with the diesel-methanol blends. [25]

3.9.2 Fuel Properties

Three fractions of diesel-methanol blends were designated for the study. Due to the low solubility of methanol in diesel fuel, a solvent consisting of oleic acid and isobutanol was added to the diesel-methanol blends to produce stabilised diesel-methanol blends. Figure 3.39 and figure 3.40 show the mass fraction of the fuel blends and the oxygen mass fraction in the fuel blends.

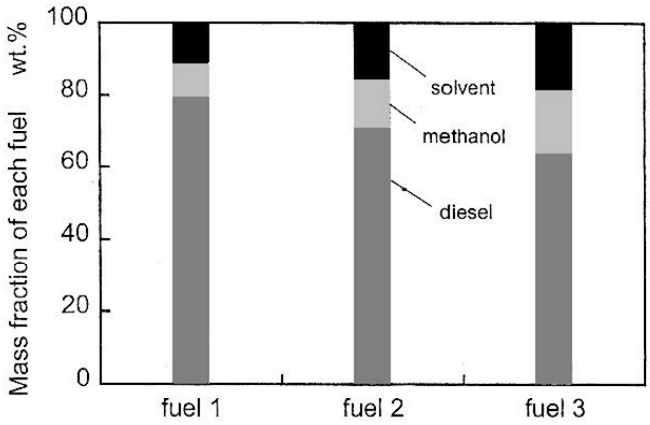

Figure 3.39: Mass fraction of the fuel blends [25]

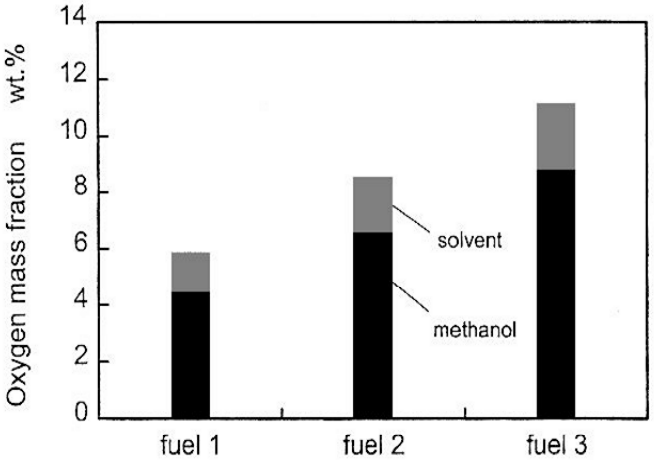

Figure 3.40: Oxygen mass fraction in the fuel blends [25]

It can be seen that the oxygen in the fuel blends mainly comes from the methanol addition although the mass fractions of methanol and solvent are the same. Therefore, it is reasonable to regard the influence of oxygen on the fuel blends as the influence of oxygen from the addition of methanol. Methanol has a high oxygen content, while the heat value and the cetane number are lower as compared to those of pure diesel fuel. For the purposes of the experiment, the three fuel blends with different methanol fractions were tested on the engine. Meanwhile, combustion characteristics and exhaust emissions were measured and analysed under the same bmep.

Tests were conducted at two engine speeds, 1500 rpm and 2000 rpm. The fuel delivery advance angle before top dead centre was – 17^{o} CA, shown by θ_{fd} in the graphs to follow.

3.9.3 Results and Analysis

The bsfc of the fuel blends are shown in figure 3.41 below.

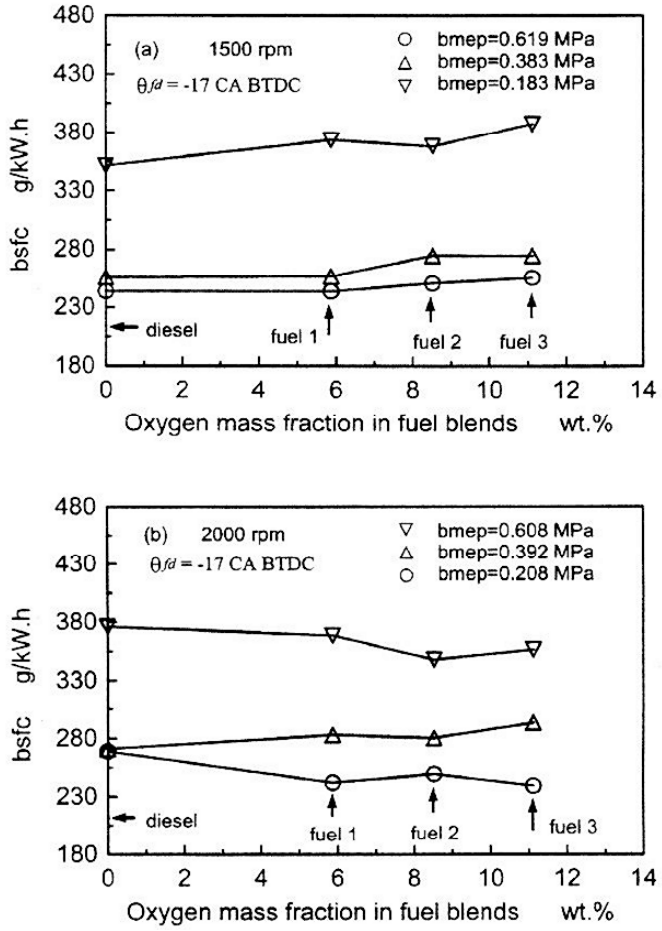

Figure 3.41: Bsfc values of the fuel blends [25]

In the case of the 1500 rpm engine speed, the bsfc increases with increasing oxygen mass fraction (or methanol mass fraction) in the fuel blends under high and low loads. It is considered that the oxygen available in the fuel blends is helpful for improving the combustion, which will benefit the reduction in the bsfc. Meanwhile, the cetane number of the diesel-methanol blends decreases with increasing methanol mass fraction in the blends, and this would cause a long ignition delay for the diesel-methanol blends, where a long ignition delay causes more fuel to be burned in the premixed burning phase and results in a high combustion pressure and an improvement in thermal efficiency.

However, more cyclic fuel is needed to produce the same bmep. Consequently, the bsfc shows an increase with increasing oxygen mass fraction. In the case of high engine speed at 2000 rpm, it was found that the bsfc did not show an increase with increasing oxygen mass fraction, however showed a slight decrease in the bsfc with increasing oxygen mass fraction. This was observed at high engine loading (bmep = 0.608 MPa). It is considered that the high swirl intensity at the high engine speed would improve the mixing process of the injected fuel, and this improvement in combustion will tend to be more obvious at the high engine load since the high combustion temperature would make the combustion of methanol more effective.

The NO_x concentrations of the fuel blends are shown in figure 3.42 below.

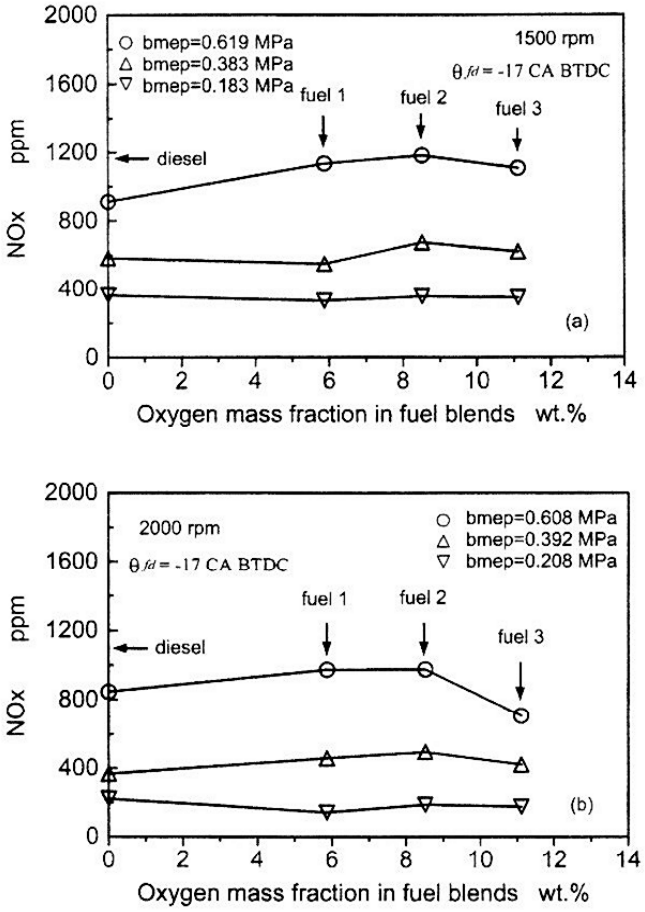

Figure 3.42: NO_x emissions of the fuel blends [25]

When the oxygen mass fraction is less than 8 wt %, the NO_x concentration increases with increasing methanol mass addition, because more fuel burns in the premixed combustion phase, where the combustion temperature is high and more oxygen is available.

However, a further increase in the methanol mass addition would accompany a high temperature drop due to an increase in the heat of evaporation of the fuel blends, furthermore an increase in the duration of fuel injection also contributes to the decrease in the NO_x concentration. The results revealed that methanol addition to diesel fuel has a strong influence on the NO_x concentration at high engine load while it has less influence at low engine load.

The CO concentrations of the fuel blends are shown in figure 3.43 below.

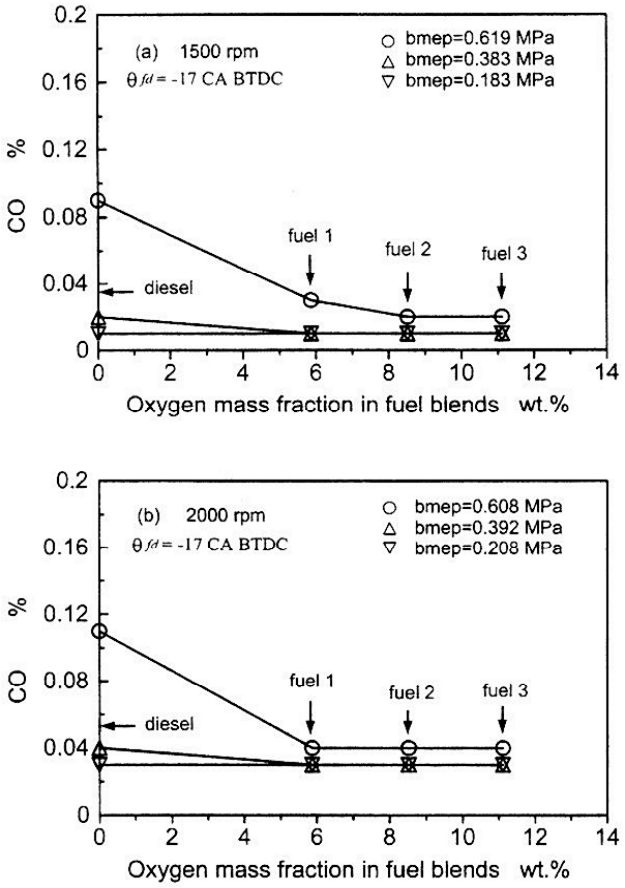

Figure 3.43: CO emissions of the fuel blends [25]

The CO concentration shows a decrease with increasing oxygen mass fraction of the diesel-methanol fuel blends, and a marked decrease in CO with increasing oxygen mass fraction present at high engine load, while a slight decrease is observed at the middle and low loads. This suggested that oxygenated fuel blends could effectively decrease the locally rich spray regions where CO is mainly formed, especially at high engine load where a larger locally rich spray region will exist due to the high swirl intensity.

In addition to this, the oxygen enrichment can also improve the post-flame oxidation of CO in the expansion and exhaust processes, and a high load results in a high cylinder temperature and consequently, a high degree of CO oxidation. The results also revealed that the addition of 6 wt % oxygen could markedly reduce the engine exhaust CO and a further increase in the oxygen mass fraction has a slight impact on the reduction of engine exhaust CO.

The HC concentrations of the fuel blends are shown in figure 3.44 below.

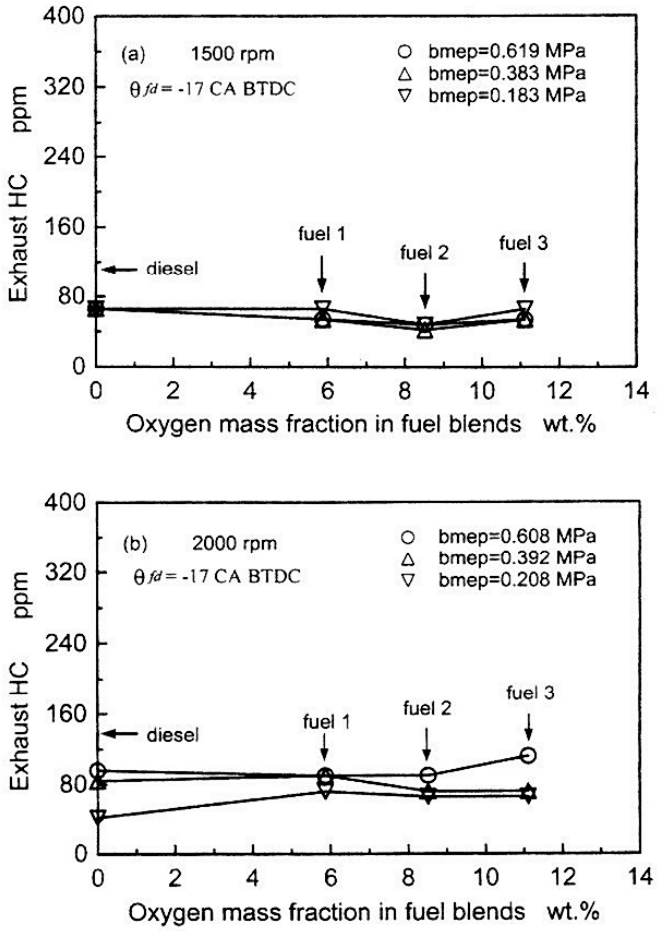

Figure 3.44: Exhaust HC emissions of the fuel blends [25]

Unlike the behaviour of both NO_x and CO versus the oxygen mass fraction of the fuel blends, the exhaust HC concentration does not vary greatly with the addition of methanol to diesel fuel, although the HC concentration shows a slight difference at different engine speeds. The exhaust HC of a compression ignition engine comes from both the rich spray region and the lean spray region.

It is suggested that the oxygenated fuel blends would have a tendency to reduce the rich spray region and to increase the lean spray region, while an increasing oxygen mass fraction would be accompanied by an increase in injected fuel and promotion of HC post-flame oxidation. Therefore, generally speaking, it can be considered that the addition of methanol to diesel fuel does not affect the exhaust HC concentrations of a compression ignition engine.

The smoke and its reduction rate in the fuel blends are shown in figure 3.45 below.

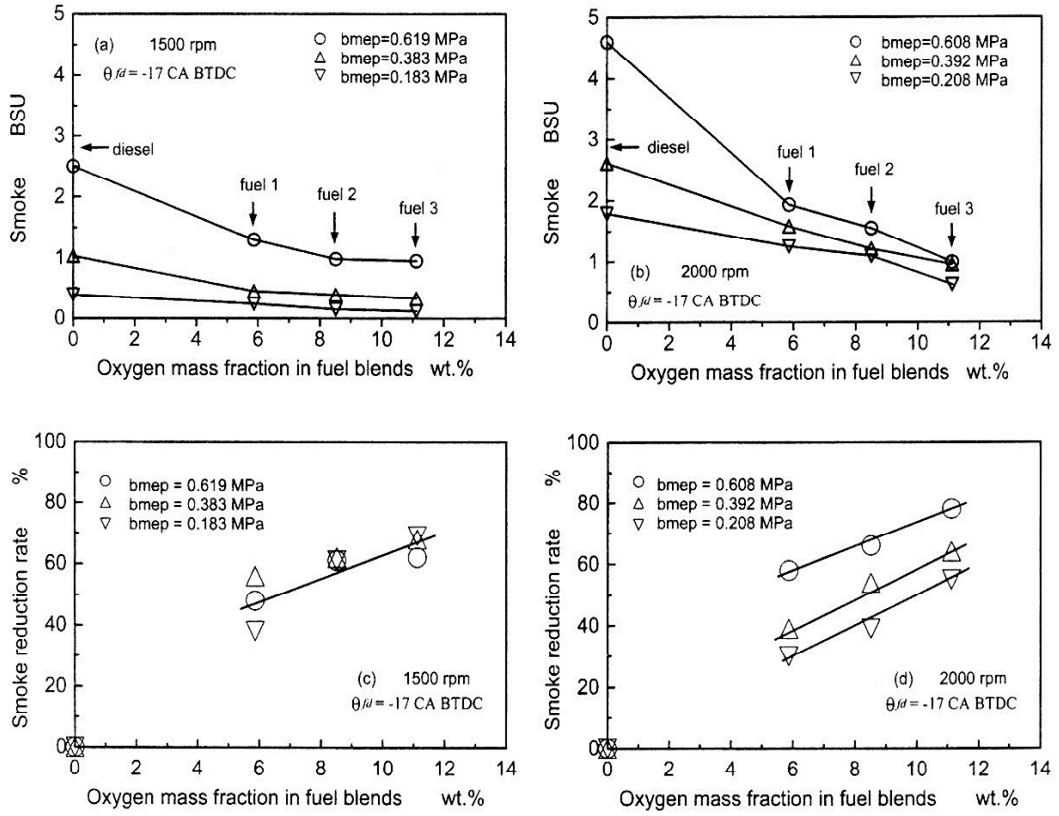

Figure 3.45: Smoke emissions and its reduction rates in the fuel blends [25]

The purpose of using oxygenated fuel blends is expected to decrease the engine smoke by supplying more oxygen to ensure the fuel burns closer to completely. The results clearly showed that the engine smoke could be decreased markedly with the addition of methanol to diesel fuel at all engine speeds and engine loads, and this is reasonable since the oxygenated fuel blends reduce the rich spray region and increase the post-flame oxidation of the formed soot. In the case of the 2000 rpm engine speed, the smoke reduction rate has a higher value at high engine load compared to that at low engine load.

In contrast to this, in the case of the 1500 rpm engine speed, the smoke reduction rate is approximately the same at low and high engine loads. Thus, it can be suggested that the high swirl intensity at high engine speeds would improve the fuel-air mixing process under high engine loads and reduce the excessive rich spray region in the combustion chamber.

The relationship between NO_x and smoke of the fuel blends are shown in figure 3.46 below.

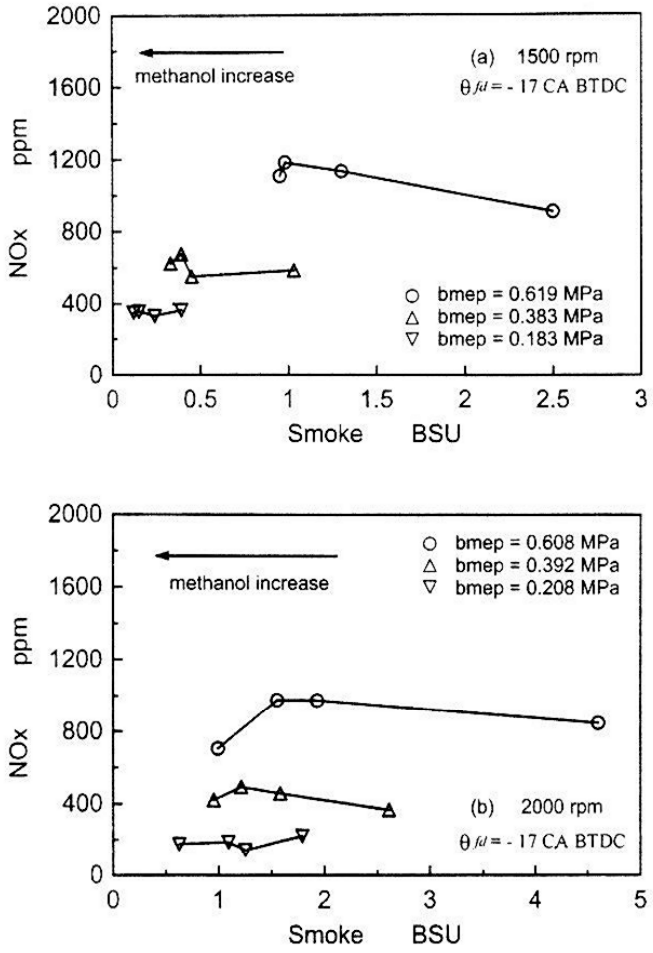

Figure 3.46: Relationship between NO_x and smoke of the fuel blends [25]

When operating with diesel-methanol blends, a flat NO_x/smoke trade off curve is presented, and an even flatter curve is observed at low engine loads. This is reasonable since the marked reduction in smoke is not accompanied by a high NO_x increase. Furthermore, a high methanol mass addition will realise the simultaneous reduction in both smoke and NO_x.

Since it is an oxygenated fuel blend, techniques such as exhaust gas recirculation can be introduced to decrease NO_x further and to realise a more pronounced reduction in both NO_x and smoke.

3.9.4 Conclusion

A stabilised diesel-methanol blend was realised and a study on the performance and emissions of the diesel-methanol blend was conducted on a compression ignition engine. The main results are summarised as follows:

i. The bsfc decreases with increasing oxygen mass fraction (or methanol mass fraction) of the diesel-methanol blends, due to an increased fraction of fuel burnt in the premixed combustion phase, oxygen enrichment and improvement in the diffusive combustion phase.

ii. Marked reductions in the exhaust CO and smoke can be achieved when operating with the diesel-methanol blend. Exhaust HC emissions do not vary much with addition of methanol to diesel fuel. However, the NO_x emissions increase with increasing methanol mass addition, and the methanol addition to diesel fuel was found to have a strong influence on NO_x concentration at high engine loads rather than at low engine loads. Lastly, a flat NO_x / smoke trade off curve existed when operating with the diesel-methanol fuel blends.

4 EXPERIMENTAL EQUIPMENT

4.1 The Compression Ignition Engine

The type of compression ignition engine used, was the Lister Petter PH2W as shown in figure 4.1. The engine is a four stroke, two cylinder, direct injection compression ignition engine. The injector pressure for the tests was set at 210 bar (21 MPa), as recommended by the manufacturer. The engine specifications are given in table 4.1.

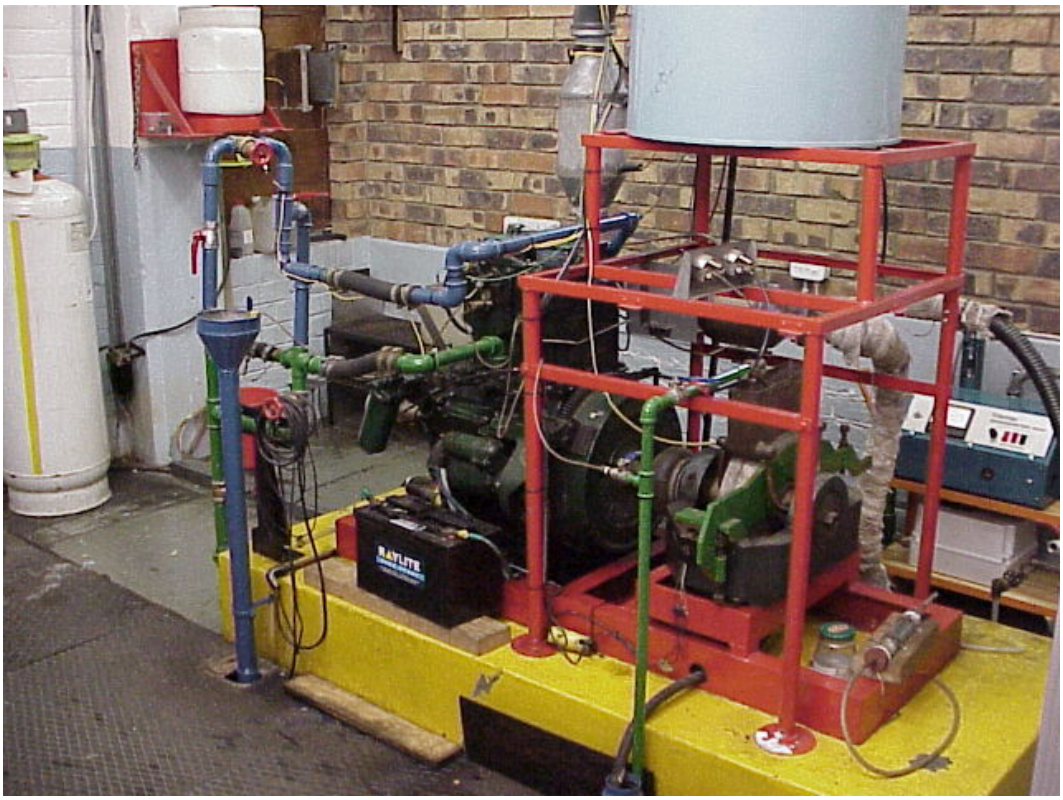

Figure 4.1: The Lister Petter PH2W Diesel Engine

The injection system shown in figure 4.2 consists of an injector pump, fuel delivery lines and the fuel injector nozzle. The injectors are supplied with fuel by two pumps, one separate pump for each injector. External pumps, for supplying the engine were not necessary as the fuel was gravity fed from fuel tanks, which were elevated two metres above the injector pumps. During motor tests the fuel flow to one cylinder stops while the flow to the other cylinder continues. Thus for motoring tests to be carried out successfully, two separate fuel injector pumps are required for independent fuel supply to each cylinder. Data can then be compared for the two cylinders, however testing of this kind was not conducted for this research. The injectors used were standard three hole type and four hole type, opening at a constant injector pressure of 210 bar for all tests. For safety reasons, usually a governor is installed onto the engine. The governor is a method of regulating the amount of fuel delivered by the fuel pump to enable a certain engine speed to be maintained. The diesel engine used, possessed no governor instead a safety switch was used, which automatically cuts the fuel supply if the engine speed exceeds 2100 rpm. The safety switch was found to be more flexible than a governor, as it could be easily set to cut out at any speed. A throttle was used to regulate the amount of fuel entering the engine by varying the travel of the pump plunger.

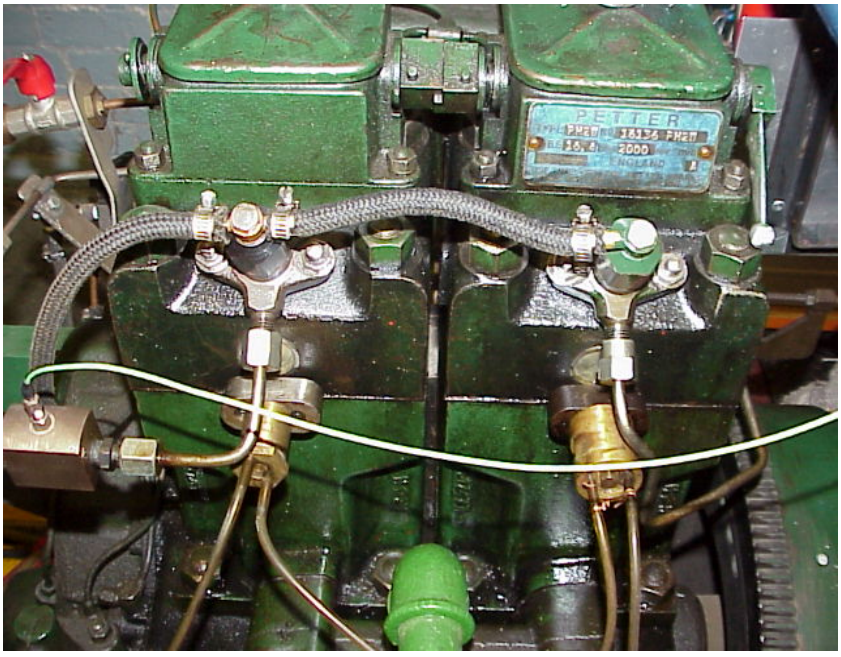

Figure 4.2: The Fuel injectors and Injector pumps

The engine was set for optimum performance on diesel fuel, thus the injection timing and injector opening pressure were adjusted for diesel fuel operation, these settings remained unaltered when the engine was fuelled on ethanol. Also included on the engine that was not tested before was the catalytic converter supplied by Magneti Marelli as shown in figure 4.3. The catalytic converter had two valves connected to it, one valve before and the other after the catalytic converter, to allow emission sampling before and after the catalytic converter.

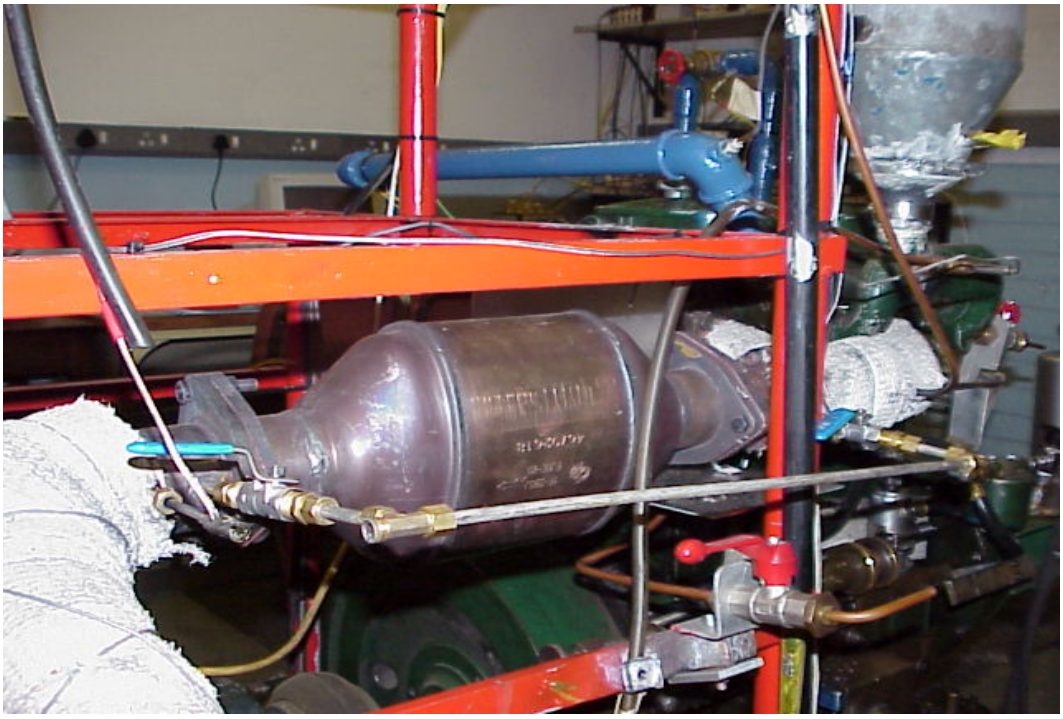

Figure 4.3: The Catalytic Converter

Table 4.1: Diesel Engine Specifications

ENGINE	
Make	Lister Petter
Bore	87.74 mm
Stroke	110.00 mm
Displacement	1330 cm^3
Compression Ratio	16.5:1
Maximum Power	13.4 kW @ 2200 rpm
Maximum Torque	66.0 Nm @ 1400 rpm
FUEL INJECTION RELEASE PRESSURES	
900 rpm – 1099 rpm	137/152 bar
1100 rpm – 2000 rpm	197/217 bar
FUEL INJECTION TIMING	
Below 1650 rpm	24^o BTDC
1651 – 2000 rpm	28^o BTDC
VALVE TIMING	
Inlet valve open	13.5^o BTDC
Inlet valve closed	38.5^o ABDC
Outlet valve open	38.5^o BBDC
Outlet valve closed	13.5^o ATDC

Table 4.2: Catalytic Converter Specifications

Make	Magneti Marelli
Type	Three-way
Monolith Material	Cordierite
Monolith Cell Density	62 per cm^2
Nominal Cell Thickness	0.15 mm
Precious Metal Ratio and Content	Pd/Rh = 5:1; $50g/ft^3$

4.2 The Fuel Supply System

The fuels tested, namely, diesel and ethanol were supplied from two separate lines. Both fuel tanks were situated at an elevated position to that of the engine. The fuel flowrate was measured directly by a flowmeter. The combustion of ethanol in compression ignition engines is enabled by ignition improvers or self-ignition promoters. Ethanol has a higher ignition temperature and lower cetane number than diesel and hence requires an ignition promoter to start ignition. The addition of a self-ignition promoter improves the ignition and combustion substantially more than other conventional means, namely by varying the injection timing or applying high compression ratios.[1]

Liquid DME stored at 4 bar in a cylinder bottle was fed to two rotameters which regulated the flowrate to each engine cylinder. The pipes connecting the cylinder to the rotameters were insulated to prevent vapourisation of DME. Vapourisation must be prevented because bubbles form in the pipes, which hampers the readings of the rotameters. The DME was fed to each engine cylinder with the delivery point being as close to the inlet valve as possible. At this point DME enters as a gas where it is mixed with air. Once the engine has warmed up, ethanol is introduced to the engine cylinder for testing.

4.2.1 Chemical Properties of Diesel, Ethanol and DME.

The chemical properties of the fuels will constantly be referred to in the discussion of the report. Only certain properties were used, however other properties have been included for information.

Table 4.3: Chemical Properties of Research Fuels [4,6,7]

Properties	Diesel	Ethanol	DME
Chemical Formula	$C_{12}H_{26}$	C_2H_5OH	CH_3OCH_3
Molecular Weight, (g/mol)	190 - 220	46.0	46.07
Carbon content, (m %)	86.0	52.2	52.2
Hydrogen content, (m %)	14.0	13.0	13.0
Oxygen content, (m %)	0	34.8	34.8
Liquid Density 25^{o}C, (kg/m^3)	840	789	668
Stoichiometric air-fuel ratio (kg/kg)	14.6	9.0	9.0
Boiling Point, (^{o}C)	180 - 360	78.32	-24.9
Autoignition Point, (^{o}C)	250	420	235
Evaporating Latent Heat, (kJ/kg)	290	904	460
Lower heating value, (MJ/kg)	42.5	26.77	28.43
Cetane Number	40 - 55	< 15	55 - 60

4.3 The Eddy Current Dynamometer

A water-cooled eddy-current dynamometer coupled to the drive shaft of the engine, was used to apply a load to the engine.

The induced load applied by the dynamometer on a load cell produced a voltage reading, which could be converted to a corresponding load in Nm. This conversion was done by using a calibration curve generated on the data acquisition system. The load applied to the engine ranged from 0 to 65 Nm and was determined by the product of the force applied to the load cell and the torque arm length.

Before testing commenced, the load cell was calibrated. Appendix A contains the calibration procedure of the load cell.

4.4 The Engine Cooling System

The cooling system used on the engine was the 'run through' water cooling system. In this system, no radiator is required because the engine is stationary, instead water is drawn from the main water supply and when it heats up it is released into the drain as waste water and replaced with new cooler water from the main water supply. A series of laboratory fans were also used to circulate cooling air in close proximity to the engine. This also assisted in maintaining correct operating temperatures. The engine temperature was maintained between 60^{o}C and 90^{o}C by varying the amount of water bled.[26]

4.5 Engine Instrumentation

4.5.1 Data Acquisition System

The high speed data acquisition system used consists of a data acquisition unit and supporting software. This system is capable of capturing substantial amounts of data at high speeds. The personal computer which was used to facilitate the software during the tests, was a Pentium II, with a 400 MHz processor, 4 GB hard drive and 64 MB RAM as shown in figure 4.4. The computer was necessary to provide the following tasks:

- Support the data acquisition system and to control the system software;
- Convert the data into comprehensive results;
- Allow the user to view the results graphically;
- Write the captured data to a hard drive for storage;

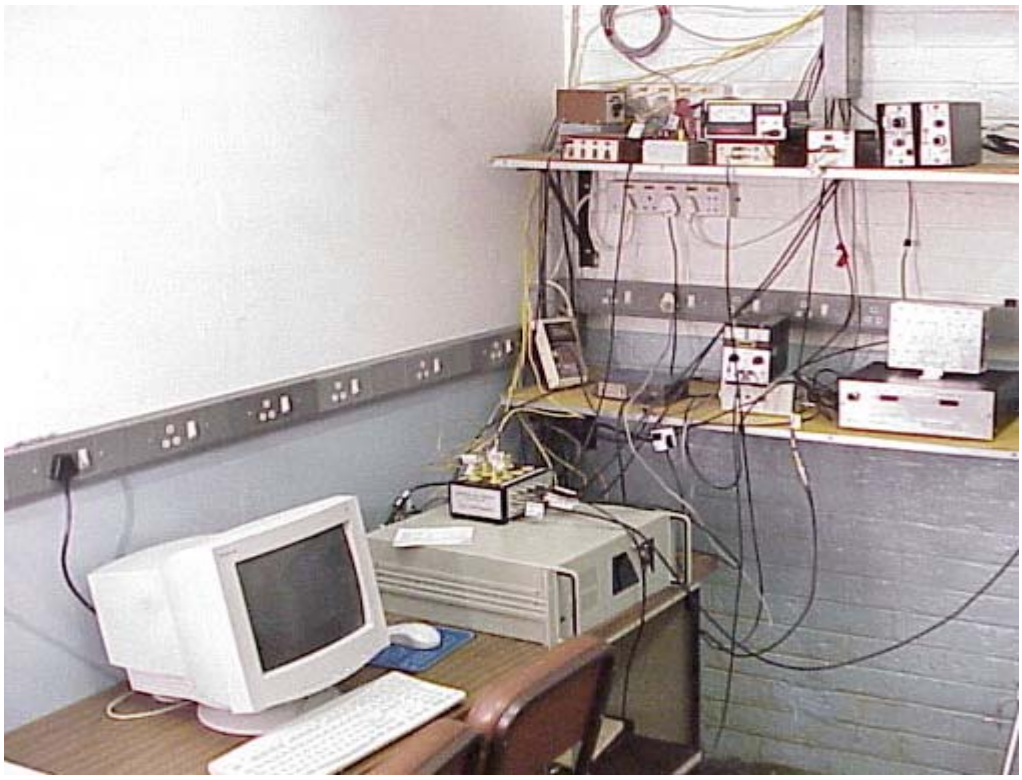

Figure 4.4: Data Acquisition System Computer

The data acquisition system captures both steady state variables and dynamic variables. The steady state channels are listed in table 4.4.

Table 4.4: Steady State Channel Number Description

Channel Number	Variable	Units	Conversion (Metric unit/Volt)	Calibration (Metric unit/Volt)
2	Fuel Flow	g/s	0.637	-0.02
3	Barometric Pressure	bar	1	0
4	Air Flow	mmH_2O	20.234	1.138
5	Engine Speed	rpm	297.77	2.5
7	Load	Nm	12.611	41.444
9	Ambient Temp.	^oC	99.6	2.273
10	Inlet Cat. Temp.	^oC	99.611	0.119
11	Outlet Cat. Temp.	^oC	98.884	3.614
12	Air Inlet Temp.	^oC	98.995	3.881
13	Inlet Water Temp.	^oC	102.29	5.869
14	Outlet Water Temp.	^oC	102.41	2.751

The system also has six dynamic channels, each of which have separate analogue-to-digital converter (ADC) cards. Here the inputs are voltages.

Data is read when the unit is set off or "triggered". This can be done internally by a clock, or externally by the TDC mark from the crank angle (CA) and TDC member unit (figure 4.5) fitted to the front shaft of the engine. Once triggered, the data would be captured at every 0.2^o CA. The system is triggered when the engine speed is within specified limits of a set value. The data acquisition system has a sampling rate of upto 1.25 MHz on the dynamic channels. This is a higher than required sampling rate.

The dynamic channels are listed in table 4.5.

Table 4.5: Dynamic Channel Number Description

Channel Number	Description
1	Trigger
2	Injector Pressure
3	Cylinder Pressure
4	Not used
5	Not used
6	Top Dead Centre (TDC)

The data acquisition system can be set for a custom refresh rate, that it stores data for a specified time before dumping it and updating it. If the system is triggered, it will assign the latest stored data to that test number. The dynamic channels are connected in parallel, allowing all dynamic channels to be triggered simultaneously. The support software performs conversion of steady state data from input voltages to the correct values by using calibration and conversion factors given in table 4.4 as the data is collected. During calibration the steady state data can be obtained from the software package as a voltage output, which is then used to generate a calibration curve.

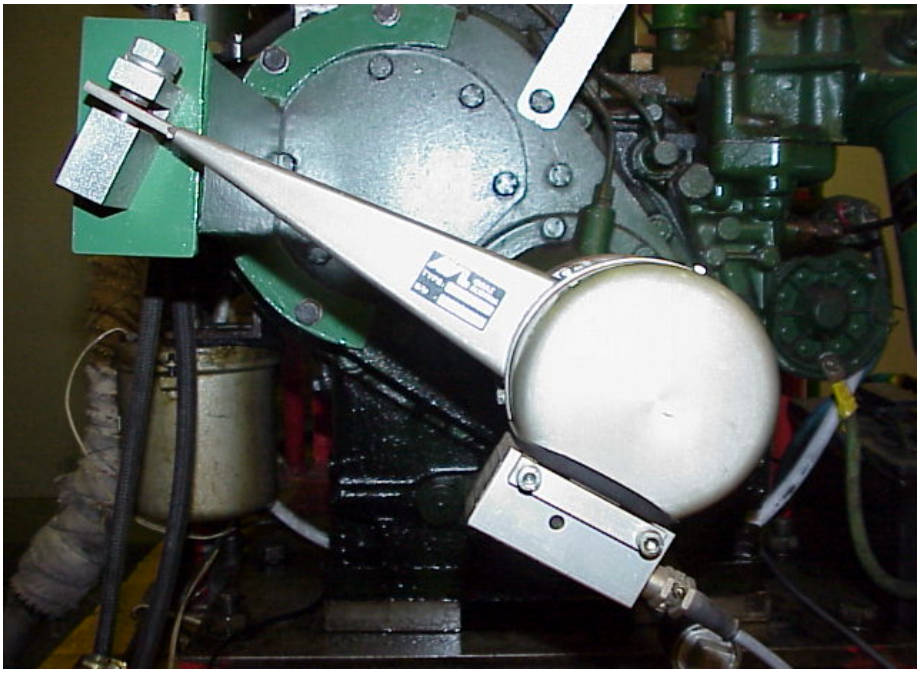

Figure 4.5: The AVL Pulse Multiplier

4.5.2 Steady State Variables

The steady state variables listed and tabulated in table 4.4 were calibrated to obtain conversion equations before testing commenced. These equations were entered into the data acquisition system as separate channels as shown in table 4.4. Temperatures measured at various locations on the engine used K-type thermocouples, which were also assigned separate channels in the data acquisition system.

Ambient Temperature

This temperature is required to calculate the air flowrate into the engine and is therefore recorded for every test. A K-type thermocouple is used and is connected to one of the data acquisition system's steady state channels.

Exhaust Temperature

The exhaust temperature was measured at various locations. One thermocouple measured the exhaust temperature in the manifold. Another thermocouple measured before catalytic converter temperature. The last thermocouple measured after catalytic converter temperature. K-type thermocouples were installed to monitor these temperatures.

Air Inlet Temperature

A K-type thermocouple was installed in the air intake drum, where the airflow orifice was located. The thermocouple was connected to the data acquisition system via a steady state channel.

Cooling Water Temperature

Both the cooling water inlet temperature and the outlet temperature were recorded also using K-type thermocouples.

Torque

The torque was measured using a load cell located under the torque arm of the eddy current dynamometer. The load cell was also connected to the data acquisition system, to one of the steady state channels.

Primary Fuel Flow Apparatus

The fuel flowrate was measured using a Pierburg flowmeter. The output of the flowmeter was first amplified and then monitored on one of the steady state channels.

Barometric Pressure

The barometer consisted of a pressure transducer with a built-in amplifier. The barometer was connected to a steady state channel and had a sensitivity of 1 mbar/mV.

Airflow

The airflow was measured using an orifice plate situated above the test rig. The orifice was made according to BS1042, which had an internal diameter of 30 mm and was mounted at one end of a 569 mm internal diameter drum. Pressure tappings were located downstream of the plate. A digital micromanometer was used to measure the pressure difference across the orifice plate. The measurement units were in mmH_2O. The output signal from the micromanometer was connected to the data acquisition system.

Engine Speed

The engine speed was measured in revolutions per minute by means of a digital tachometer that was connected to the AVL Pulse Multiplier (figure 4.5). This was then connected to the crank angle encoder. The output from the tachometer was also connected to a steady state channel.

4.5.3 Dynamic Variables

Injector Pressure

The injector pressure was measured by means of a pressure transducer located in the fuel line between the injector pump and the injector. The output from the transducer was fed to a charge amplifier having a transducer sensitivity of 2.5 and a conversion factor of 50 mechanical units per volt. The output from the amplifier was connected to the data acquisition system to a dynamic input channel.

Cylinder Pressure

The cylinder pressure of the number one cylinder of the engine was taken. A high-pressure piezo-electric pressure transducer was used to take measurements. A passage was drilled through the cylinder head and a brass sleeve was fitted into the passage. The pressure transducer was situated inside the brass sleeve. The brass sleeve had a water jacket through which cold water was flowing to keep the transducer cool. The output from the transducer was then fed to a charge amplifier having a transducer sensitivity of 15 and a conversion factor of 20 mechanical units per volt. The output from the amplifier was then connected to the data acquisition system via a dynamic input channel.

Top Dead Centre (TDC)

TDC was measured using a signal sent from the AVL pulse multiplier, when the TDC mark was detected on the crank angle encoder. The signal was used to trigger the data acquisition system every 0.2^{o} CA, as well as to indicate the position of TDC in each of the cycles recorded. The TDC pulse was connected to the trigger as well as to one of the dynamic channels.

4.6 The Emissions Analysis Equipment

4.6.1 Signal Gas Analyser

The emissions analyser consists of four separate units to measure CO_2, CO, NO_x and THC. The system also includes a gas cooler drier unit and a filter-oven unit to remove water vapour from the exhaust gas. The complete system is shown in figure 4.6.

Figure 4.6: The Signal Gas Analyser

A heated line is used to convey the samples from the exhaust to the filter unit. From the filter unit, part of the sample is heated before being introduced to the NO_x and THC analysers. The remaining portion of sample gas is sent to the cooler drier unit before being introduced to the CO_2 and CO analysers.

The CO_2 and CO analysers are non-dispersive infra-red analysers, NDIR's which selectively absorbs infra-red radiation over a narrow range of wavelengths. The gas to be examined is drawn through a measuring cell in which pulses of broad-band infra-red light is passed. The degree of absorption of energy in the relevant wave band is measured by a system of detector cells and gives a measure of the concentration of the corresponding gas in the sample.[2]

The NO_x analyser uses a chemiluminescence detector, CLD. In this instrument, use is made of the chemical reaction between nitric oxide and ozone:

$$NO + O_3 \rightarrow NO_2 + O_2 \rightarrow NO_2 + O_2 + \text{photon}$$

The nitrogen compounds in the exhaust gas are a mixture of NO and NO_2 described as NO_x. In the detector the NO_2 is first catalytically converted to NO and the sample is reacted with ozone, generated by an electric discharge through oxygen, at low pressure in a heated vacuum chamber. The light emitted in the above reaction is measured by a photomultiplier and indicates the NO_x concentration in the sample.[2]

The THC analyser uses a flame ionisation detector, FID. The operation of this instrument depends on the production of free electrons and positive ions that takes place during the combustion of hydrocarbons. If the combustion is arranged to take place in an electric field, the current flow between anode and cathode is closely proportional to the number of carbon atoms taking part in the production reaction. The corresponding unburned hydrocarbons level is thus to some extent affected by the hydrogen and carbon ratio of the fuel. In the detector, the sample is mixed with hydrogen and helium, and burned in a chamber, which is heated to prevent condensation of the water vapour formed.[2]

The analysers are to be switched on one hour before the testing to attain operating conditions. The general start up procedure for the gas analyser unit is listed in Appendix B. The output of the analysers was monitored in real time on a personal computer.

4.6.2 The Hartridge Smoke Meter

The Hartridge Smoke Meter shown in figure 4.7 was used to measure the opacity of the exhaust gas smoke, in Hartridge Smoke units (HSU). The meter was connected to the exhaust system of the engine by means of a rubber hose. Smoke readings were taken after the data for an individual test has been recorded, so as not to interfere with the analyser flow requirements.

Figure 4.7: The Hartridge Smoke Meter

4.6.3 The Oxygen Meter

The percentage of oxygen present in the exhaust emissions was measured using the Servomex Oxygen Analyser (type OA 250) shown in figure 4.8. This instrument provides an analogue scale reading of the percentage of the oxygen present in the exhaust gases. The sample is fed through a filter and then through copper sulphate ($CuSO_4$) crystals, so as to remove any moisture in the emissions before being introduced to the oxygen meter.

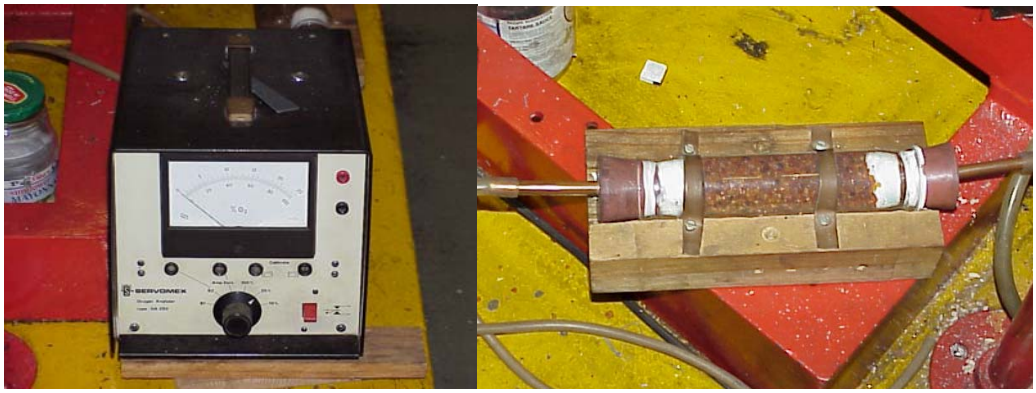

Figure 4.8: The Oxygen Meter and Filter Crystals

5 EXPERIMENTAL PROCEDURE

Tests were first performed using diesel fuel and then repeated with ethanol/DME at the same speed. For both the diesel and ethanol tests, a standard three hole injector was used. However, for the ethanol fuelling, additional tests were performed with a standard four hole injector. In all the tests the same injector opening pressure was maintained. The engine speed at which the tests were conducted was 1550 rpm and the loads applied ranged from 0 to 65 Nm.

The procedure below was used for all the readings and necessary precautions were also adhered to.

5.1 Testing Procedure

- Switch on the emissions analyser unit at least an hour before testing, to ensure all the analysers reach the correct operating temperatures before calibration could be performed. The calibration procedures are given in Appendix B.
- Switch on all the other instrumentation. Turn on all the computers and open the relevant test software. Note that instrumentation should be turned on before the computer software is opened, as the software locates input signals from the data capturing equipment, prior to testing.
- Connect the oxygen analyser to the copper sulphate crystals filter.
- Check all instrumentation and connections, and start the engine on pure dimethyl ether or diesel depending on which tests are conducted first. Run the engine at low load for a few minutes to warm up. While this is happening, check all instruments for correct operation and when the engine reaches the correct operating conditions, save a sample test to check that data is saved correctly.

- It is very important to monitor the cylinder pressure trace on the computer throughout the testing. The transducer is prone to clogging and then delivers inaccurate pressure traces to the acquisition system. Inaccuracies in pressure readings render the testing useless because calculations within the software are performed with pressure as the primary variable.
- The acquisition program records data such as fuel flow, airflow, ambient temperature and load. A useful manual calculation is that of the DME flowrate as a percentage of total fuel. Annotate the rotameter reading and calculate the corresponding mass flowrate and express the DME flowrate as a percentage of total fuel flowrate. The rotameter reading is manually recorded for each test.
- When the defined speed range is reached, the program will capture data and provide the user the option of accepting or rejecting the test. Depending on a number of factors such as fuel flow reading, actual speed rpm and maximum cylinder pressure, the operator will evaluate the worth of the data, and if satisfied, will save the test. The test data is written to the pre-defined test data directory under a filename that is automatically incremented for each test.
- As a test is accepted, the computer program for emissions is also initiated. A standard emissions test takes 100 readings at intervals for one second, for each species. In other words, an emissions test takes 100 seconds to complete. During any test at least two emissions tests are recorded.
- For each load setting of the dynamometer, at least four sets of engine data are recorded. Approximately 10 load settings are taken for both the diesel and ethanol/DME tests. Loads above 65 Nm with diesel fuelling usually result in the engine being over-fuelled resulting in excessive black smoke being emitted.
- Once testing is complete, the engine is shut down and the purge and shutdown procedures for the emissions analyser are followed.

5.2 Precautions

Due to the nature of the equipment being used, certain precautions have to be taken to ensure safety, reliability of the equipment and accuracy of the results.

5.2.1 Engine

- Ensure the battery is charged and correctly connected.
- Ensure the injector pressures are set correctly.
- Ensure that the fuel injectors are tightened before each test to prevent leakage.
- Ensure the oil level is correct and changed when necessary.
- Check the water supply to the cylinder pressure transducers.
- During operation, keep away from all moving parts.
- Ensure sufficient ventilation of exhaust gases from the test cell.
- Wear hearing protectors due to engine noise.

5.2.2 Fuel System

- Ensure that ice water and cloths are readily available before testing.
- Check the rotameters during testing for any blockages in the rotameter measurement glass.
- Check for leaks, in the DME line, while the engine is idling.
- Ensure that there is sufficient fuel available in the fuel tanks.

5.2.3 Dynamometer

- Ensure water supply is on.
- Beware of rotating parts.
- Beware of electric shock.

5.2.4 Emissions Analyser

- Ensure sufficient calibration gases are available.
- Ensure the filters of the individual analysers are regularly inspected and serviced to ensure the sufficient flow of the sample gases.
- Ensure the calibration procedures are followed correctly before each test session.
- Ensure the analysers are allowed to purge after each test session.

6 RESULTS

The results tabulated in appendix C are divided into three sections. These sections are performance, emissions and catalytic converter efficiency. All tests were conducted at a speed of 1550 rpm and at an injector opening pressure of 210 bar. A number of tests were conducted at each load setting, however the best values for a particular test were selected to best depict the trend observed for the test in question. Reference should be made to appendices D and E for the complete results of the tests conducted. Attention should be rendered to the headings of the fields of the tables in appendix C, as these are the only aspects discussed. The graphs to be presented in chapter 7 were plotted from the data tabulated.

7 DISCUSSION

The discussion consists of three sections, namely, performance, emissions and catalytic converter efficiency. Diesel and ethanol/DME fuel trends with a three-hole injector and ethanol/DME with a four-hole injector are compared and discussed. Due to the nature of research work, the anticipated discussion of the results obtained can only be verified further with continued research into diesel engine performance and emissions, equipped with a catalytic converter.

7.1 Performance

The parameters which will be discussed are brake power, brake specific fuel consumption, fuel conversion efficiency, ignition delay, combustion chamber maximum pressure, mechanical efficiency, and the temperature before the catalytic converter (i.e. the exhaust temperature). These parameters are very important as they dictate how the engine performs and significant comparisons can be made between diesel and ethanol fuel. The settings of the engine during this research, i.e. the injector pressure and injector timing, were set for optimum performance on diesel fuel and remained unaltered when the engine was fuelled on ethanol/DME fuel. This may have an impact on the performance and emissions results of the engine when fuelled on ethanol as this may not necessarily be optimum for ethanol fuel.

7.1.1 Brake Power

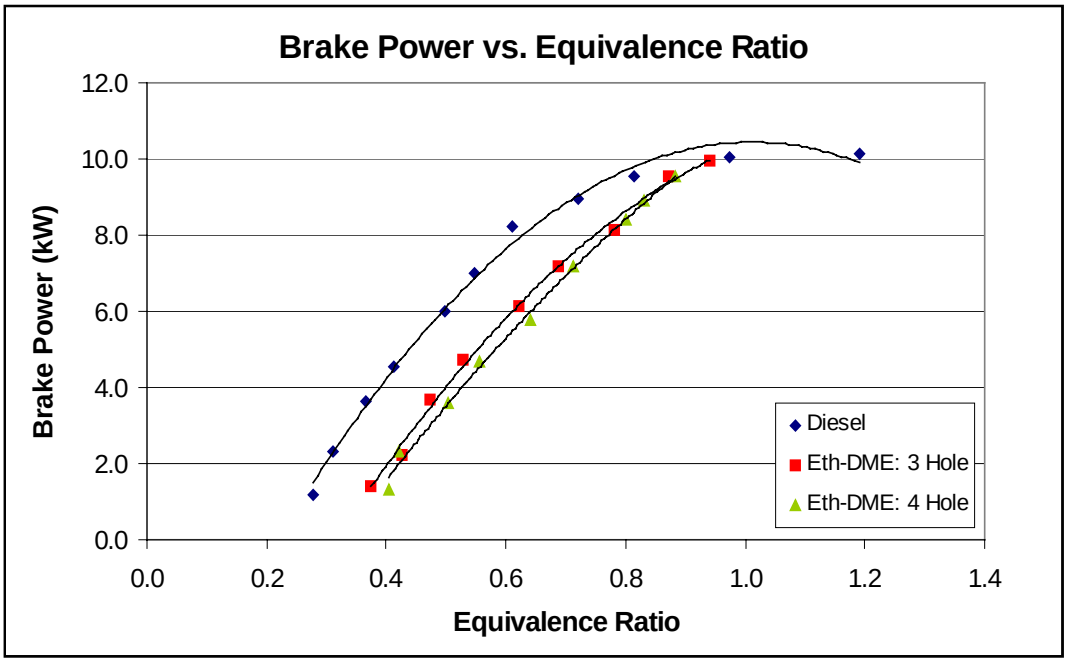

Figure 7.1: Brake Power vs. Equivalence Ratio

Figure 7.1 shows the comparison of the brake power output of the engine against equivalence ratio, ϕ. Brake power is defined as the usable power dissipated by the engine to overcome variable loading applied to the engine. Fuel/air equivalence ratio is the ratio of the actual fuel/air mass flowrate ratio to the stoichiometric ratio, defined as $\phi = \frac{\left(\dot{m}_f / \dot{m}_a\right)_{actual}}{\left(\dot{m}_f / \dot{m}_a\right)_{stoichiometry}}$, where $\dot{m}$ is mass flowrate[1]. An equivalence ratio, $\phi < 1$ denotes a lean mixture, indicating that more air is available than needed for total combustion, $\phi > 1$ denotes a rich mixture, meaning there is an over supply of fuel. For $\phi = 1$, there exists stoichiometric proportions of fuel and air, i.e. there is just sufficient oxygen for the conversion of all fuel into completely oxidized products[1]. An expected trend is observed, the brake power increases with an increasing equivalence ratio. The equivalence ratio for diesel fuel extends from approximately 0.27 to 1.2, while that of ethanol/DME for both injector configurations extends from approximately 0.38 to 0.94.

The maximum brake power attained by diesel and ethanol/DME with the 3 hole injector is in the range of 10 kW and 10.2 kW, as compared to ethanol/DME with the 4 hole injector only reaching approximately 9.5 kW.

As can be seen from the curves, to obtain the same power output ethanol/DME requires more fuel than diesel, because for the same power a correspondingly higher equivalence ratio exists for ethanol/DME.

This is to be expected, since the calorific values for both ethanol and DME are 26.77 MJ/kg and 28.43 MJ/kg respectively, which is approximately 65% that of diesel fuel at 42.5 MJ/kg (Table 4.3).

As can be seen from the curves, there is great similarity between ethanol/DME 3 hole and 4 hole configurations as ϕ increases, however the 3 hole outputs slightly more power when ϕ approaches closer to 1. However, if the engine setting were changed to better suit ethanol/DME fuel, it may well be possible to use less fuel and achieve lower ϕ values.

For all tests, had more fuel been introduced the power would begin to deteriorate and the engine would no longer operate efficiently, especially in the case of diesel fuelling, where excessive black smoke is produced.

7.1.2 Brake Specific Fuel Consumption

Fuel consumption is measured as a flowrate - mass flow per unit time, $\dot{m}_f$. A more useful parameter is the brake specific fuel consumption, bsfc, defined as the fuel flow per unit power output[1]. The power output referred to is the total power from the gas within the cylinder to the piston and also the power absorbed in overcoming engine friction and driving engine accessories. The equation for bsfc is shown below.

$$bsfc = \frac{\dot{m}_f}{Power_b}\ ^{[1]} \qquad [7.1.1]$$

Figure 7.2 shows the comparisons of the brake specific consumption against brake power.

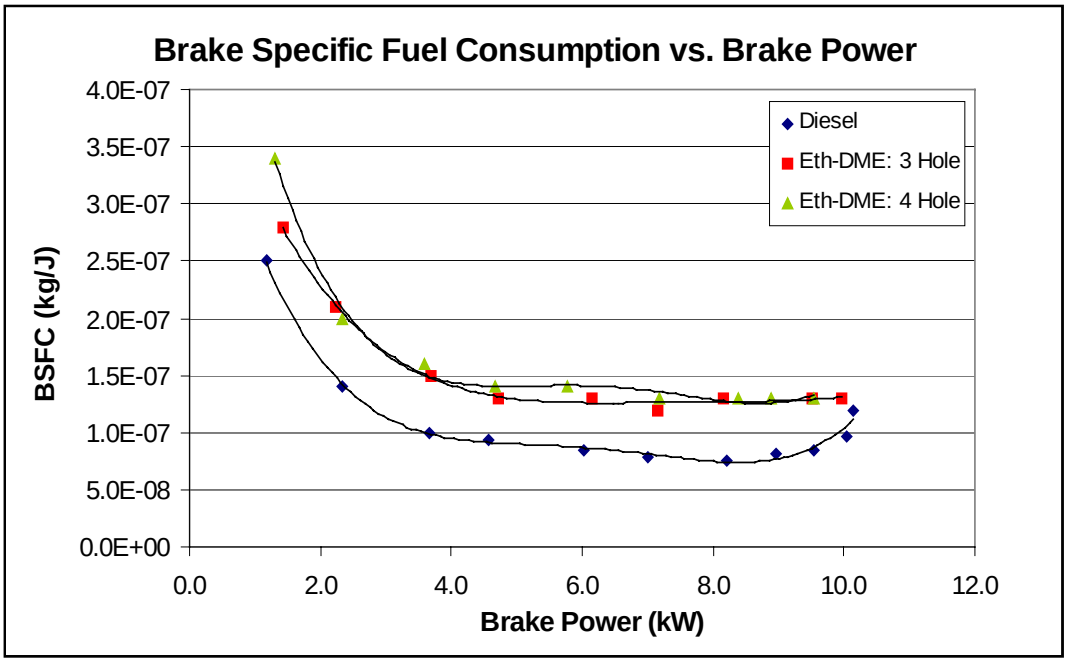

Figure 7.2: Brake Specific Fuel Consumption vs. Brake Power

Diesel and ethanol/DME possess similar trends, the bsfc is high for low loads and decreases progressively as brake power increases. This is expected, as the combustion chamber approaches optimum operating conditions with respect to combustion chamber pressures and temperatures.

As mentioned previously, more ethanol/DME fuel is required to produce the same power output and bmep as diesel fuel. As depicted, for the load range from 1.15 kW to 9.5 kW, the average ratio of bsfc of ethanol/DME fuel to diesel fuel is approximately 1.5. This is due to ethanol/DME's lower calorific value, however for higher loads, above a power output of 9.5 kW, diesel bsfc increases reaching ethanol/DME's bsfc. This sudden rise confirms that over-fuelling does indeed occur as brake power output approaches 10 kW, as previously shown and discussed in figure 7.1. The difference between the 3 hole and 4 hole ethanol/DME nozzles is very modest, however there may exist a worthy difference in fuel conversion efficiencies, which is discussed next.

7.1.3 Fuel Conversion Efficiency

A dimensionless parameter that relates the desired engine output (work per cycle or power) to the necessary input (fuel flow), is the ratio of the work produced per cycle to the amount of fuel energy supplied per cycle, that can be released in the combustion process. The fuel energy supplied, which can be released by combustion, is given by the mass of fuel supplied to the engine per cycle times the heating value of the fuel. The heating value of a fuel, Q_{HV}, defines its energy content. It is a measure of the engine's efficiency, which is called the fuel conversion efficiency, η_f.

$$\eta_f = \frac{Power}{\dot{m}_f Q_{HV}} \quad [1] \qquad [7.1.2]$$

Fuel conversion efficiency and brake specific fuel conversion consumption, defined by equations 7.1.2 and 7.1.1 respectively, are inversely proportional in relation to one another.

Figure 7.3 shows the comparisons of the fuel conversion efficiencies against ϕ.

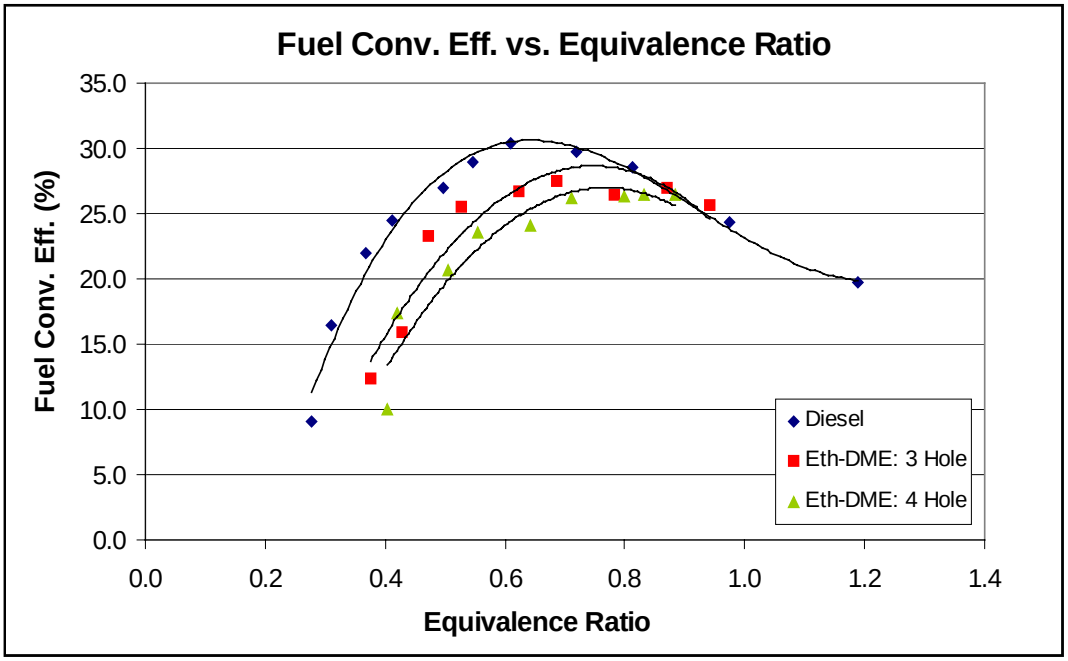

Figure 7.3: Fuel Conversion Efficiency vs. Equivalence Ratio

Diesel fuelling is more efficient than ethanol/DME, reaching a maximum of approximately 31%, at an equivalence ratio of 0.63.

Ethanol/DME 3 hole reaches a maximum of approximately 28% at an equivalence ratio of 0.69 and ethanol/DME 4 hole reaches a maximum of approximately 26.5% at an equivalence ratio of 0.83.

The trend is that the fuel conversion efficiency increases, peaks and then decreases with increasing equivalence ratio for the three curves.

A reason for the lower efficiency of ethanol/DME fuelling could be due to the aspiration of DME during air intake, and possibly less air being aspirated into the cylinder. This amount of air could be insufficient to oxidise the fuel and facilitate combustion. In the case of diesel fuelling, only air is aspirated, maximum efficiency occurs at lower equivalence ratios, this shows that more air is available for combustion, resulting in a higher efficiency. When comparing the 3 hole and 4 hole configurations for ethanol/DME fuelling, the 3 hole achieves a higher efficiency, because with the 4 hole configuration more fuel is present in the mixture. Hence the amount of air available for combustion is less.

In addition to the amount of air aspirated for combustion, the amount of work done between the two fuels is equally important. Ethanol/DME fuelling releases energy during the compression stroke before top dead centre (TDC), which results in negative work being done as compared to diesel. Diesel releases energy just before TDC and during the expansion stroke, resulting in minimal negative work being done.[1] To explain this in greater depth, the pressure traces of the fuels will be analysed.

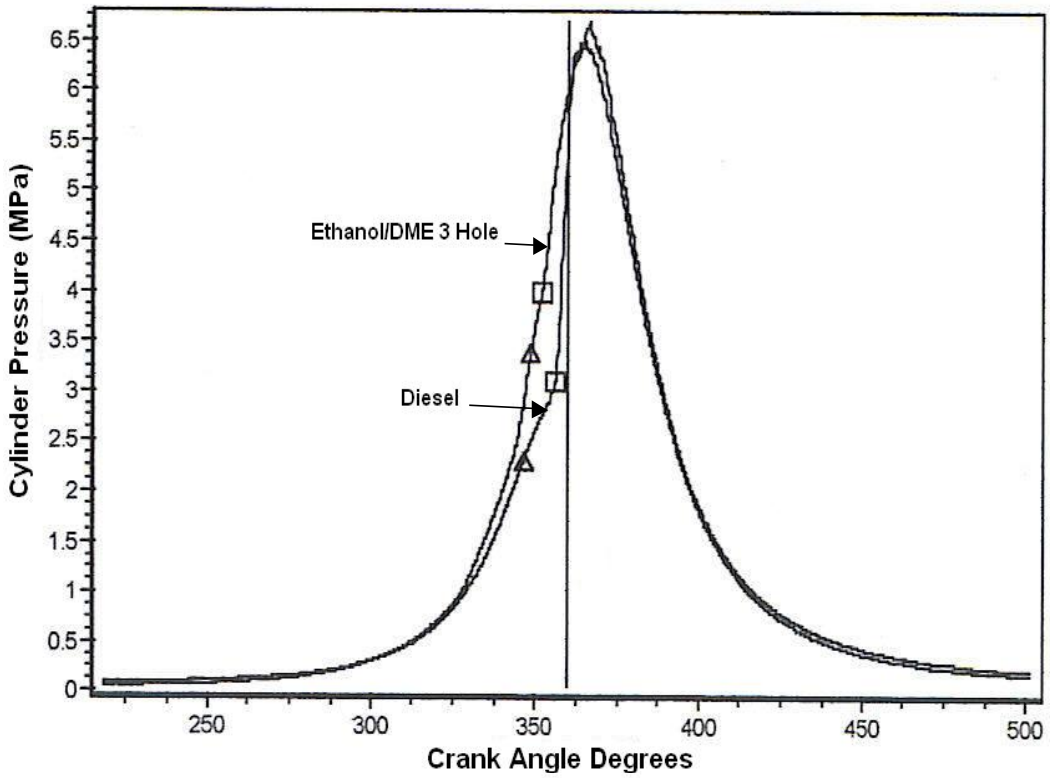

Figure 7.4: Cylinder Pressure Trace vs. Crank Angle – Diesel vs. Ethanol/DME

For diesel, as shown in figure 7.4, the pressure increases gradually until fuel is injected into the cylinder. The injection point is shown by a triangular symbol. The pressure continues to rise until the ignition point is reached, shown by a rectangular symbol, just before TDC. TDC is shown by the vertical line midpoint of the CA degree axis.

Immediately after the ignition point, the sudden steep and rapid pressure rise indicates the amount of energy released during the four-stroke cycle. For the ethanol/DME 3 hole trace, the pressure rise is much smoother than the diesel trace, however the ignition point of ethanol/DME is a few CA degrees sooner than the ignition point of diesel. The relative positions of these ignition points to TDC is important in quantifying the amount of negative work done by the piston during the cycle. Amounting from this is the greater area enclosed between the pressure trace, TDC and the ignition point of ethanol/DME when compared to diesel. This difference in areas results in the improved efficiency values attained for diesel combustion. Ideally the ignition point should coincide with TDC for maximum fuel conversion efficiency.

When comparing the ethanol/DME 3 hole and 4 hole efficiencies, the 3 hole achieves higher efficiencies throughout the load range. Figure 7.5 shows the pressure trace of ethanol/DME for the two configurations.

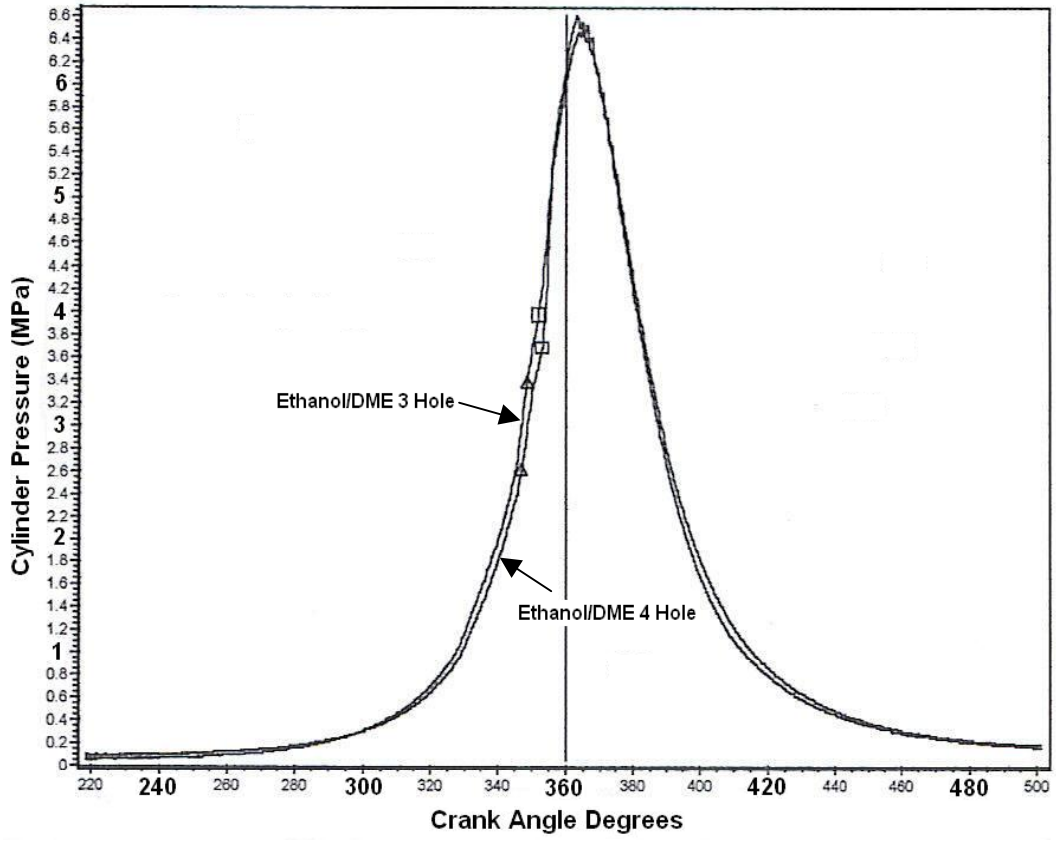

Figure 7.5: Cylinder Pressure Trace vs. Crank Angle – Ethanol/DME 3 hole vs. Ethanol/DME 4 hole

The traces are very similar in nature, however the injection and ignition points occur at different pressures. For the 3 hole, the injection and ignition points occur at higher pressures than the 4 hole points. Although the CA degrees are essentially the same for the ignition points of the 3 hole and 4 hole, the area enclosed between the pressure trace, TDC and the ignition point for the 4 hole is greater. Due to this, more negative work is done hence the lower efficiencies achieved for the 4 hole injector.

7.1.4 Ignition Delay

As fuel is injected into the combustion chamber, it forms very fine droplets, vapourises and mixes with air before ignition, leading to combustion. The ignition delay period is defined as the time or crank angle degrees between start of injection and start of combustion. The delay is primarily a function of those parameters, which affect the chemical reaction rate of the fuel vapour mixtures. These parameters include pressure, temperature and entry velocity into the combustion chamber of the mixture.[7] Physical factors such as injection timing, injection quantity, combustion chamber wall effects, oxygen concentration and fuel properties, also plays a role in ignition delay. Figure 7.6 shows the change in ignition delay with increasing brake power.

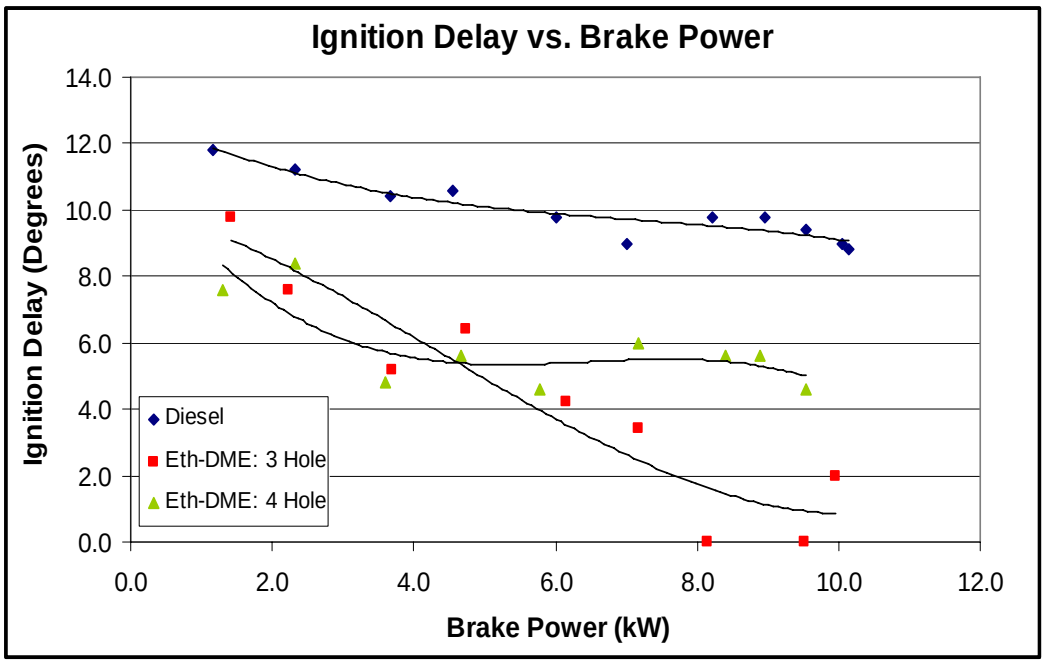

Figure 7.6: Ignition Delay vs. Brake Power

In the case of diesel fuel, the ignition delay decreases almost linearly with increasing brake power.

The ethanol fuel follows the same trend, however much lower ignition delays are achieved. This decreasing trend for both fuels is due to the increase in combustion chamber temperature. As load is increased, the residual gas temperature, as well as the chamber wall temperature increase.

This results in a higher fuel charge temperature at injection, thus shortening the ignition delay.[1] The lower ignition delay of ethanol is also affected by the energy release characteristics of DME.

DME ignites and releases energy in two stages before top dead centre, resulting in the ethanol being injected into a high temperature zone in the combustion chamber, which promotes speedy ignition and prepares a favourable environment for ethanol to burn.[27]

The differences in ignition delay between the 3 hole configuration and 4 hole configuration could be due to the combustion characteristics, as the 3 hole configuration consistently possesses higher fuel conversion efficiencies than the 4 hole configuration. Furthermore, although the same amount of air is aspirated during the intake stroke, for the 4 hole configuration more fuel is present, therefore the local amount of oxygen available for combustion is less, as previously discussed. Oxygen concentration plays an important role in ignition delay, the higher the concentration of oxygen, the lower the ignition delay, however this is outweighed by temperature and pressure effects.

Fuel properties also play a significant role in ignition delay, such as the cetane number of the fuel. The cetane number is a measure of the ability of the fuel to autoignite. The higher the cetane number, the lower ignition delay.[1,3] From the curves, it is clear that the cetane number does affect the ignition delay. Referring to Table 4.3, diesel has a cetane number in the range of 40 – 55, ethanol less than 15 and DME greater than 55. Although ethanol fuel has a cetane number lower than that of diesel, during injection, when combined with DME suggests an overall higher cetane number of the fuel mixture, hence the lower ignition delay curve.

According to Heywood[1], injector nozzle configuration, fuel atomisation, fuel spray penetration and vapourisation rate over reasonable ranges do not appear to influence the duration of the delay period significantly, especially in a warmed up engine. The differences in the ethanol ignition delays could thus only be attributed to the amount of DME aspirated into the cylinder for the two configurations.

From the results tables (Appendix D) the DME equivalence ratios of the 3 hole and 4 hole are basically the same for a brake power output of 0 to 4.6 kW, beyond which the 3 hole decreases almost linearly to an ignition delay of 0^{o} CA, as shown in figure 7.6.

For the brake power output beyond 4.6 kW, the DME equivalence ratios and the ethanol equivalence ratios for the 3 hole are higher than that of the 4 hole, reaching maximum values of 0.114 and 0.49 for the 3 hole, and 0.112 and 0.447 for the 4 hole configurations respectively. These values indicate that more DME is available for combustion in the 3 hole configuration. The 3 hole also reaches a zero ignition delay, physically meaning the fuel is ignited and burns at the nozzle tip. This physical occurrence shows the effectiveness of DME as ignition promoter, allowing ethanol to ignite almost immediately upon entering the combustion chamber.

7.1.5 Combustion Chamber Pressure

The combustion chamber pressure for both fuels follow the same trend; namely the pressure increases with increasing equivalence ratio. The maximum pressure reached is approximately 69 bar at an equivalence ratio of 0.82, as depicted in figure 7.7 below.

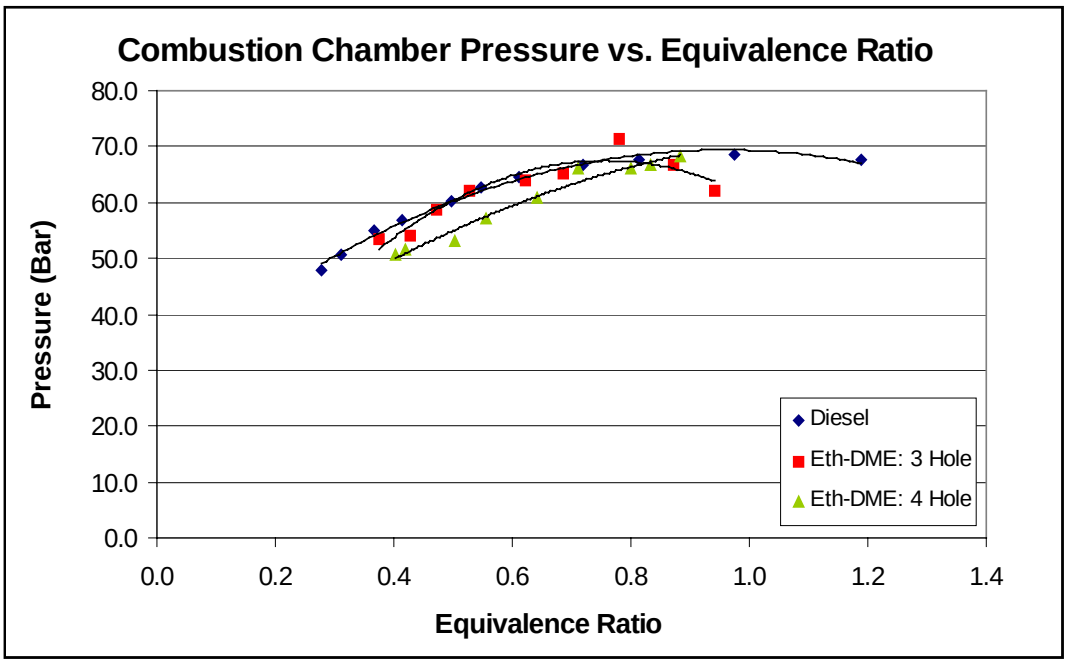

Figure 7.7: Combustion Chamber Pressure vs. Equivalence Ratio

However, there is a difference between the two fuels and the two injector configurations at lower equivalence ratios. Figure 7.8 shows an expanded pressure axis scale to depict the differences in maximum pressure reached between the two fuels and two configurations.

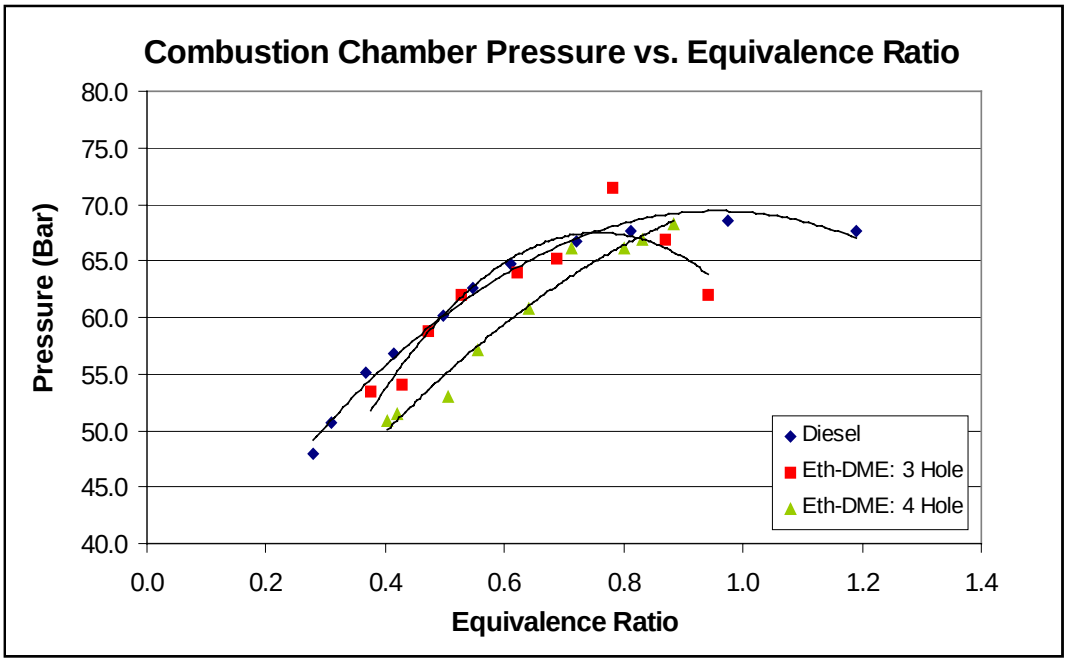

Figure 7.8: Combustion Chamber Pressure vs. Equivalence Ratio - Expanded

For the same pressure value, diesel has the lowest equivalence ratio followed by ethanol 3 hole and lastly the ethanol 4 hole configuration. As equivalence ratio increases, the difference between the pressure values progressively decrease until maximum pressure is attained, afterwhich pressure decreases for equivalence ratios above stoichiometry (ϕ=1).

The reason for this phenomenon is due to the enthalpies of vapourisation of the two fuels. The enthalpy of vapourisation is a good measure of the strength of the intermolecular forces that hold the molecules together in a liquid.[28] The enthalpy of vapourisation of ethanol is high hence its poor ignition quality. As a result of this poor ignition characteristic of ethanol, the combustion pressure rise is dependent and limited by the ignition and energy release of DME during ignition delay. Diesel has a much lower enthalpy of vapourisation and superior fuel conversion efficiency, hence reaches a higher pressure at a lower equivalence ratio. The fuel conversion efficiency of ethanol 3 hole and 4 hole approached that of diesel, this was also discussed in figure 7.3, as the equivalence ratio approached 0.8, hence similar pressures were reached between the two fuels. However, interesting to note, the trend for ethanol/DME may well posses higher pressures had further tests been conducted for equivalence ratios greater than 0.85.

7.1.6 Before Catalytic Converter Temperature

The temperature before the catalytic converter is actually the exhaust temperature in the manifold after combustion. This temperature is an important factor in the performance analysis of the catalytic converter. The temperature was noted to increase with increasing brake power as illustrated below in figure 7.9.

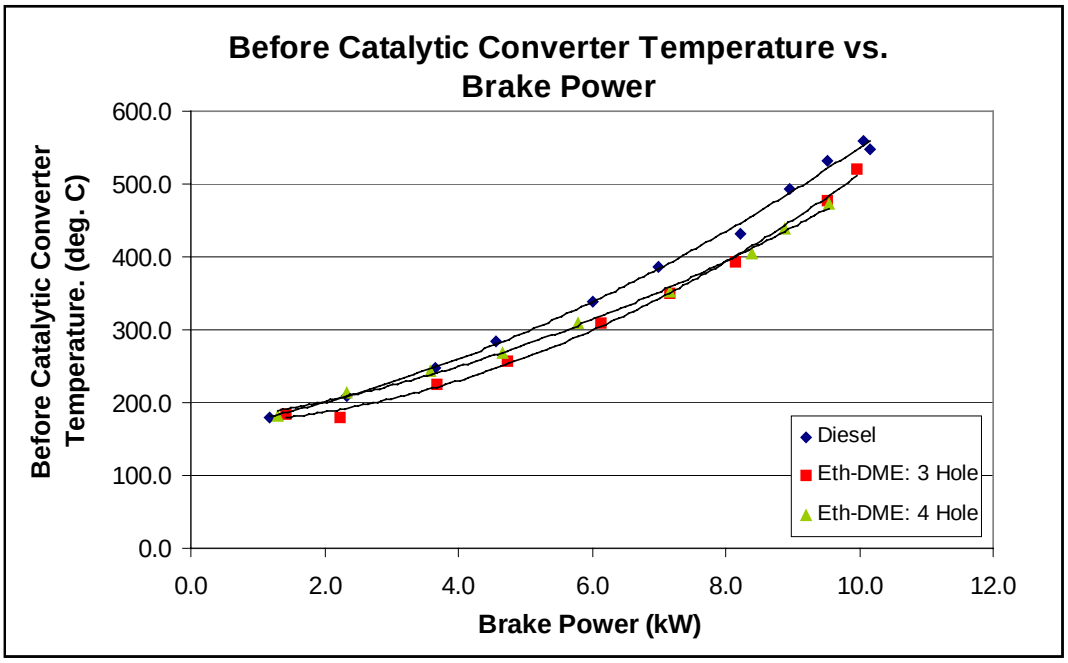

Figure 7.9: Before Catalytic Converter Temperature vs. Brake Power

The diesel fuel and ethanol/DME fuel show the same trend, however diesel fuel has slightly higher temperatures as compared to both 3 hole and 4 hole ethanol/DME fuelling. As discussed previously, the pressure attained by diesel fuel is higher and from the fundamental gas law, temperature is directly proportional to pressure, hence the higher temperature of diesel. The maximum temperature reached by diesel is approximately 550 ^{o}C and by ethanol/DME 3 hole and 4 hole is approximately 520 ^{o}C. The temperature increase is also due to the increase of fuel being burnt in the combustion chamber for increasing load.

The lower temperatures for ethanol/DME have been shown by Cipolat[19] to be due to the energy release pattern of DME. Cipolat[19] has shown that DME releases energy in two stages.

The first stage releases an approximately constant amount, while the second stage releases an increasing amount with increasing equivalence ratio, which in turn increases with brake power. This second stage release of energy is thus the balance of the energy in the DME. Diesel fuel releases energy in a single stage.

The slight difference in temperature between the 4 hole and 3 hole configurations of ethanol fuel could be due to the amount of fuel which impinges on the wall of the combustion chamber. The 4 hole configuration provides a 140^o angle of dispersion from the nozzle during injection as compared to the 3 hole, which provides a 130^o angle of dispersion. Hence, the 10^o difference existing results in more fuel impinging onto the chamber wall circumference for the 4 hole configuration. Combustion chamber temperature is also related to the amount of fuel that is physically on the wall during burning. Also, residual energy from the ignition of DME will render higher temperatures on the walls. This seems to be the case for brake power outputs greater than 8 kW, where ignition delay for the 3 hole configuration tends to zero, hence the slightly higher temperatures for the 3 hole configuration in this mentioned range. Heywood[1] also confirms that lower number of nozzle holes result in lower burning rates of the fuel, this affects temperature differentials obtained in the combustion chamber between the two nozzle configurations.

7.1.7 Mechanical Efficiency

Mechanical efficiency is the ratio of brake power delivered by the engine to indicated power. The equations[1] below illustrate this relationship.

$$\eta_m = \frac{P_b}{P_{ig}} = 1 - \frac{P_f}{P_{ig}} \qquad [7.1.3]$$

and

$$P_{ig} = P_b + P_f \,. \qquad [7.1.4]$$

Where η_m is the mechanical efficiency, P_{ig} is the gross indicated power, P_b is the brake power and P_f is the friction power. Indicated power is the rate of work transfer from the gas mixture within the cylinder to the piston. Brake power is the power absorbed in overcoming engine friction, driving engine accessories and pumping power. Friction power is power used to overcome the friction of the bearings, pistons and other mechanical components of the engine and to drive the engine accessories.[1]

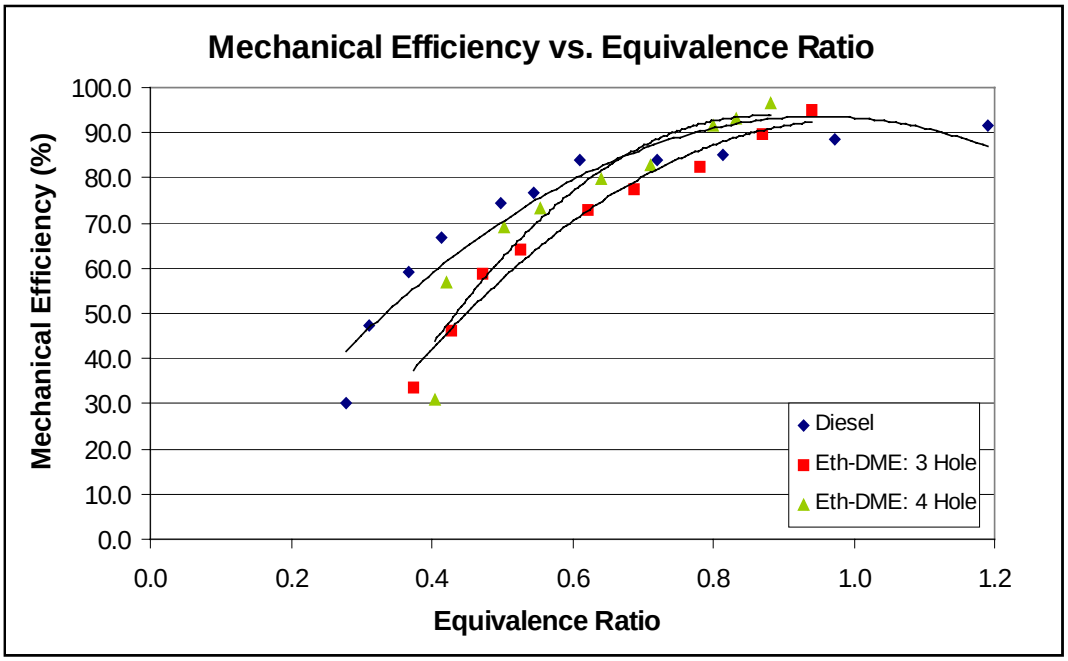

Figure 7.10: Mechanical Efficiency vs. Equivalence Ratio

Figure 7.10 shows a similar trend between the two fuels as the equivalence ratio increases. Ethanol fuel 3 hole achieves the lowest efficiencies throughout the load range, however ethanol 4 hole achieves comparable efficiencies to that of diesel fuel when approaching stoichiometry.

The process which plays a significant role in mechanical efficiency is the conversion of chemical energy into useful mechanical energy. This conversion in turn relates to the rapid combustion of fuel within the combustion chamber. Ignition delay, combustion chamber temperature and pressure, facilitates this conversion process. These parameters could attribute to ethanol 4 hole's comparable mechanical efficiency to that of diesel fuel.

7.2 Emissions

The emission species which will be discussed are nitrogen oxide, total hydrocarbons, carbon monoxide, carbon dioxide, oxygen and smoke. The emission species were monitored before (upstream) and after (downstream) the catalytic converter, during testing. Comparisons will first be made before the catalytic converter and subsequently after the catalytic converter. The discussion of the species will take the form of specific emissions, defined as the concentration per unit power output. The concentrations of species in the engine exhaust are measured in parts per million by volume and normalised by dividing by the power output. Normalised indicators of emission species are far more useful in making direct comparisons between the fuels. The independent variable against which the specific emission is compared, is equivalence ratio. Engine power and equivalence ratio are considered important in the analysis of emissions.

Analysis before the catalytic converter

7.2.1 Specific NO_x emissions

Nitrogen oxides (NO_x) are formed throughout the combustion chamber due to the reactions of atomic oxygen and nitrogen. The process of formation is highly temperature dependent and hence also pressure dependent, however the latter being dependent on a secondary level. NO_x emissions scale proportionally to the engine load and are relatively low during engine start and warm up. The results obtained confirms the preceding observation by Ferguson[6], where the concentration of NO_x increases for increasing brake power. Figure 7.11 shows the trends for the fuels.

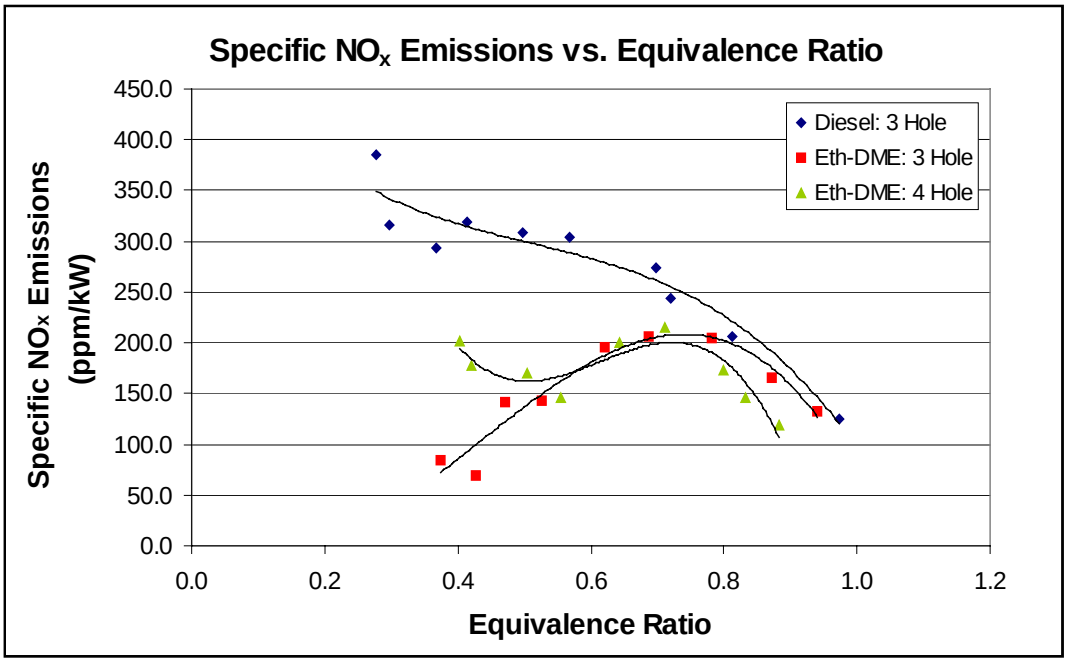

Figure 7.11: Specific NO_x emissions vs. Equivalence Ratio before Catalytic Converter

Diesel fuel possesses an almost linear decrease in specific emissions, ethanol/DME 3 hole possesses a negative quadratic trend and ethanol/DME 4 hole a negative cubic trend, all observed for an increase in equivalence ratio. Heywood[1] has established the equation that governs these phenomena is given by:

$$\frac{d[NO]}{dt} = \frac{6\times10^{16}}{T^{0.5}}\exp\left(\frac{-69.090}{T}\right)[O_2]_e^{0.5}[N_2]_e \quad \text{mol/cm}^3\text{.s} \qquad [7.2.1]$$

where [ ] denotes species concentrations in mol/cm^3 and subscript e, equilibrium temperature concentrations. Although the above equation provides for the calculation of the concentration of NO, NO_x is mostly comprised of NO. Temperature dependency is also highlighted in the equation above and this is due to the activation energies of the extended Zeldovich mechanism, as described in the literature survey. Therefore, NO_x chemical rates are non-linear functions of temperature[6], hence the decrease phenomena even though combustion chamber temperature increases. However, added to this is the NO_x emission and power ratio, produced at each brake power output. It is clear that the very dissimilar trends shown in figure 7.11 between diesel and ethanol/DME up to an equivalence ratio of approximately 0.75, is mainly due to the concentration of NO_x produced as opposed to the corresponding power output divisor.

Ethanol/DME 3 hole and 4 hole are very dissimilar up to an equivalence ratio of 0.6, once again due to the higher NO_x emissions produced for the 4 hole configuration nozzle.

NO_x formation also depends on the amount of oxygen consumed during combustion. Oxygen atoms dissociate and attach to nitrogen atoms to form nitric oxide. A good indication of the amount of oxygen in the combustion chamber is given by the equivalence ratio.

For an equivalence ratio above approximately 0.75, the decrease in specific NO_x is mainly due to a lack of oxygen (as found in a rich mixture) during combustion, for all three trends shown in figure 7.11. Milton[3] suggests that the difference between diesel and ethanol/DME fuels could be attributed to the fuel properties such as the latent heat requirements of the additional fuel in the rich mixtures. Differences in fuel cetane numbers also have an effect on the formation of NO, a decrease in cetane number would result in an increase in NO formation which is mainly attributed to the effect of the pre-ignition radicals on the mechanism of NO formation.[8] The difference between the 4 hole and the 3 hole ethanol/DME fuel is due to the slightly lower temperatures attained for the 3 hole configuration, as shown in figure 7.9, especially for low ϕ values. As previously discussed in figure 7.9, the 3 hole configuration temperatures are slightly higher than the 4 hole for a brake power output of 8 kW and above. Therefore, the deduction could be drawn that for an equivalence ratio above approximately 0.8, the temperature mechanism dominates the oxygen content in the fuel charge, for the formation of NO_x.

7.2.2 Specific THC emissions

Total hydrocarbon (THC) emissions result from the presence of unburned fuel in the exhaust of an engine. Some of the exhaust hydrocarbons are not found in the parent fuel, but are hydrocarbons derived from fuel whose structure is altered within the cylinder by chemical reactions that are incomplete. These are about 50 % of THC's.[6] There exists six mechanisms which are responsible for the alternate oxidation pathways and exhaust THC appearing – namely crevices; oil layers; carbon deposits; liquid fuel; cylinder wall flame quenching and exhaust valve leakage. Of these mentioned, the crevice mechanism is the most significant in producing THC emissions, accounting for 40 % of the total.[29] During compression and early stages of combustion, the cylinder pressure rises forcing a small fraction of fuel-air mixture into the crevices of the combustion chamber wall.

The crevice temperatures are approximately equal to the cooled wall temperatures, hence the density of the fuel-air mixture in the crevices is greater than in the cylinder. When the cylinder pressure decreases during the latter portion of the expansion stroke, the unburned crevice gases return to the cylinder resulting in hydrocarbon emissions.[6]

The region referred to, experiences rapidly decreasing temperature and pressure, and is called a quench zone, most of the THC initially exist here. Quench zones can basically be defined as the region where a flame cannot be supported.[3]

Figure 7.12 depicts the specific THC emissions with increasing equivalence ratio.

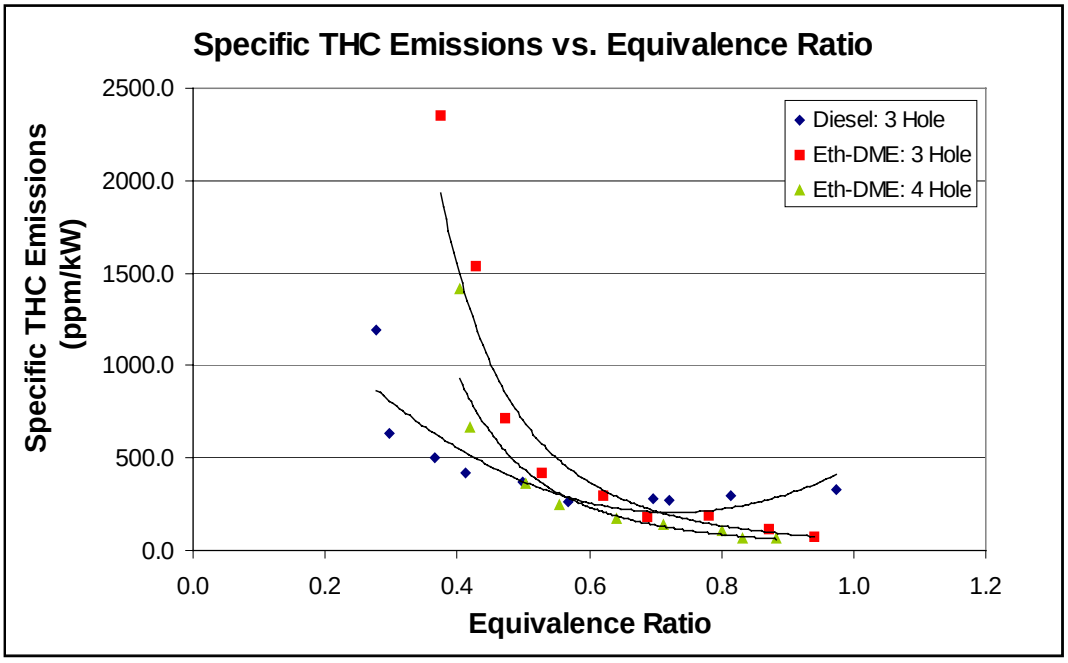

Figure 7.12: Specific THC emissions vs. Equivalence Ratio before Catalytic Converter

For diesel, THC concentration levels decrease steadily, reach a minimum then steadily increase as the equivalence ratio increases. For ethanol/DME, the concentration is high for low equivalence ratios, then decreases sharply before leveling off as equivalence ratio approaches unity. The ethanol/DME 3 hole and 4 hole possess a similarly decreasing trend as diesel below an approximate equivalence ratio of 0.7. For values greater than this equivalence ratio, diesel and ethanol/DME show opposing trends.

The ethanol/DME trends could largely be attributed to the energy release characteristics of DME, which promotes ignition and combustion of the ethanol fuel introduced in the latter portion of injection. The elevated temperature of the combustion chamber allows for a more effective combustion of ethanol, which in turn results in a reduced quenching effect in the crevices of the combustion chamber, as confirmed by the ignition delay results previously discussed.

Due to the crevice mechanism being the dominant cause of THC, physical properties such as fuel viscosity and density between diesel and ethanol/DME could be an additional reason for the difference between the two fuels. Diesel fuel is more viscous than ethanol/DME[1,26], hence accessibility to the fuel mixture in the crevices, during combustion is more difficult with diesel fuel. The chemical structure difference between diesel and ethanol/DME is another possible reason for the difference in trends between the two fuels. Diesel is solely composed of carbon and hydrocarbon atoms as compared to ethanol/DME, which also contains oxygen atoms[26,30]. Due to ethanol/DME containing oxygen atoms, 34.7% weight[30], sufficient combustion for lower THC is achieved more readily as compared to diesel fuel. Carbon and hydrogen atoms content are considerably higher in diesel, hence the tendency of higher concentrations of THC to form.

The difference in the 3 hole and 4 hole ethanol/DME configurations is due to the spray characteristics of the fuel during injection. With the 4 hole configuration, more fuel is present in the fuel jet, meaning that the relative amount of oxygen atoms, by virtue of ethanol/DME's chemical structure, in the 4 hole configuration is slightly more than the 3 hole configuration. Overlean regions in the fuel jet could also result in THC in that the fuel mixes beyond the combustible lean limit where combustion cannot be sustained. The higher THC emissions for the 3 hole is also based on the assumption that these overlean regions are slightly more pronounced as compared to the 4 hole configuration.

7.2.3 Specific CO emissions

Carbon monoxide is formed during intermediate combustion stages of hydrocarbon fuels. The amount of CO formed has been found to be fundamentally related to the equivalence ratio[3]. Heywood[1] has found that for fuel-rich mixtures, CO concentrations in the exhaust increase steadily with increasing equivalence ratio, and for fuel-lean mixtures, CO concentrations in the exhaust vary less with the equivalence ratio when compared to fuel-rich mixtures, this is especially the case for diesel fuel.

Diesel engines operate well on the lean side of stoichiometry, hence CO emissions are low and not a practical problem, however for the purposes of this research these emissions will be analysed.

Figure 7.13 shows the variation of specific emissions of CO for an increase in equivalence ratio.

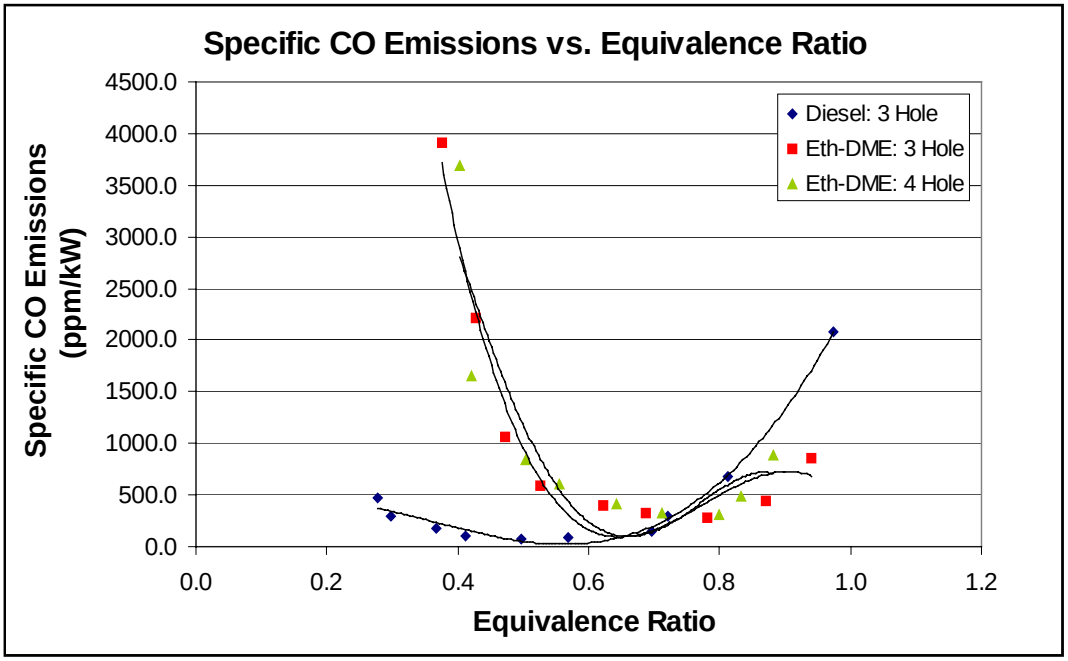

Figure 7.13: Specific CO emissions vs. Equivalence Ratio before Catalytic Converter

Diesel fuel specific CO emissions initially decrease and reach a minimum for equivalence ratios below 0.63, afterwhich it resembles an exponential trend. Specific CO emissions resulting from ethanol/DME are substantially higher at lower equivalence ratios. As equivalence ratio increases, due to an increase in load, the specific CO emissions for ethanol/DME decrease sharply and reach a minimum at an equivalence ratio of approximately 0.7. Beyond this, ethanol/DME increases nearly in the same proportions as diesel, however, close to stoichiometry, diesel specific CO is more than double that of ethanol/DME 3 hole and 4 hole. CO emissions are dependent on the amount of oxygen available to complete combustion. The higher concentrations of ethanol/DME at lower values of ϕ or lower brake power output, is due to the DME being aspirated during intake resulting in less air being available than diesel to complete combustion (Appendix D). Egnell[31] suggests that this phenomenon is due to the worse mixing between the fuel and air, which results in a lower local oxygen concentration and thus decreased CO oxidation.

The minimums reached for both fuels in figure 7.13 above, correspond to the maximum fuel conversion efficiencies achieved, as discussed in figure 7.3 earlier.

The difference between diesel and ethanol/DME could be largely attributed to chemical kinetics, with ethanol/DME containing more oxygen molecules to support dissociation of CO, leading to near complete combustion.[3]

In addition to this is the absence of carbon to carbon bonds and the lower C/H ratio of ethanol and DME, giving lower emissions of CO.[31]

The slight difference between the 3 hole and 4 hole ethanol/DME configurations could be due to the fuel-air mixing in the fuel sprays from the nozzles. The equivalence ratios are more for the 4 hole configuration corresponding to the same specific CO emissions as the 3 hole, thus meaning more fuel is present in the mixture.

7.2.4 Specific CO_2 emissions

The combustion of carbon and carbon-containing compounds in excess oxygen leads to the formation of carbon dioxide, CO_2. Due to the presence of hydrocarbon chains in diesel and ethanol/DME, CO_2 formation does occur. The mechanism by which CO_2 is formed, is through the oxidation of CO. CO formation is one of the principal reaction steps in the hydrocarbon combustion mechanism, which may be summarised as follows:

$$RH \rightarrow R \rightarrow RO_2 \rightarrow RCHO \rightarrow RCO \rightarrow CO \quad [1] \qquad [7.2.2]$$

where R stands for the hydrocarbon radical. The CO formed in the combustion process (when the products of combustion cool during expansion within the cylinder)[3] via this path is then oxidised to CO_2, at a slower rate. The principal CO oxidation reaction in the hydrocarbon air flames arising in the premixed combustion phase is as follows:

$$CO + OH \rightarrow CO_2 + H \quad [1] \qquad [7.2.3]$$

In the premixed hydrocarbon air flames, the CO concentration increases rapidly in the flame zone to a maximum value.[1] Thus the CO_2 emissions produced is closely related to the amount of CO in the exhaust gases. The specific CO_2 emissions of the two fuels follow similar trends, as shown in figure 7.14.

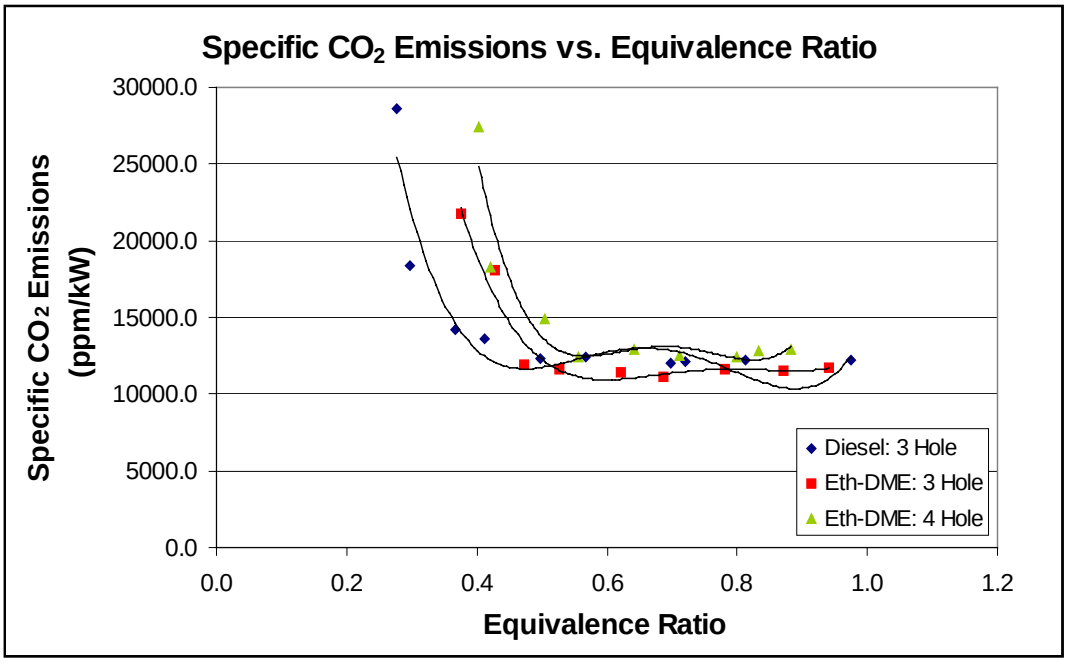

Figure 7.14: Specific CO_2 emissions vs. Equivalence Ratio before Catalytic Converter

As depicted, diesel fuel produces CO_2 emissions earlier than ethanol/DME. This could mean that during air intake the oxygen available is used for the oxidation of CO to CO_2. This could also be due to the carbon content in diesel, which is almost double that of ethanol/DME and the fact that more hydrocarbon radicals are present, as confirmed by the previously discussed THC emissions. For equivalence ratios between approximately 0.4 and 0.52, corresponding specific CO_2 emissions for ethanol/DME are considerably higher than those of diesel, which is mainly as a result of ethanol/DME being an oxygenated fuel. The difference in the 3 hole and 4 hole ethanol/DME configurations is due to the difference in CO present, with the 4 hole producing slightly more, hence the greater local formation of CO_2.

7.2.5 Specific O_2 emissions

The primary source of oxygen is from the air intake stroke into the combustion chamber. The secondary source is as a result of chemical kinetic reactions in the fuel charge during combustion.

The variation of specific O_2 emissions against equivalence ratio is shown in figure 7.15.

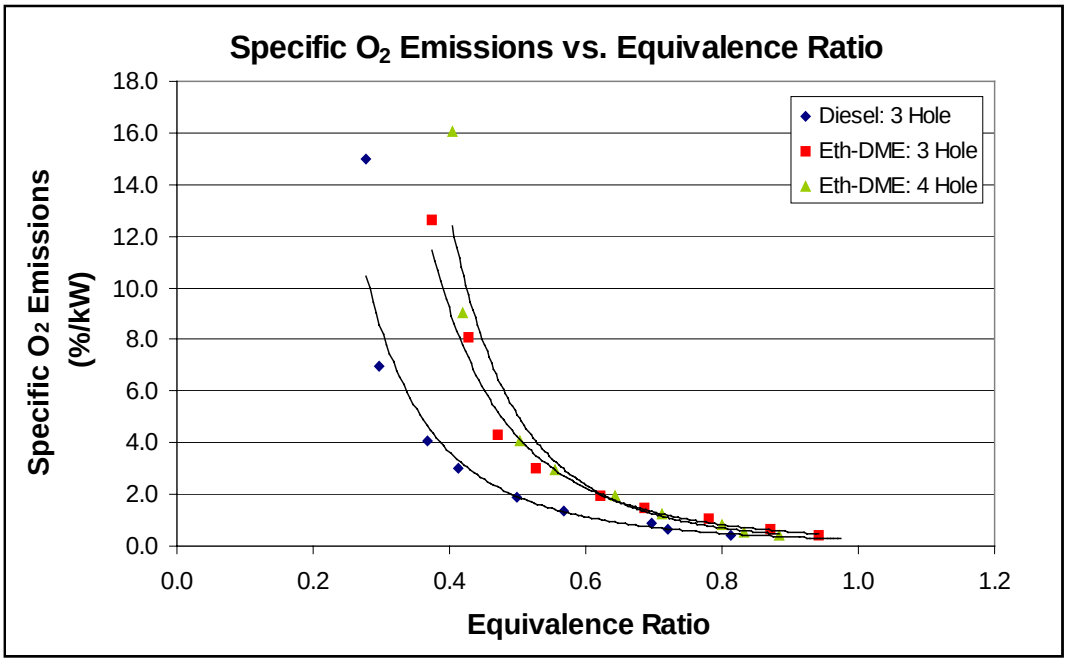

Figure 7.15: Specific O_2 emissions vs. Equivalence Ratio before Catalytic Converter

The amount of oxygen decreases sharply with increasing equivalence ratio. The trends between the two fuels are similar except, as in the case of specific CO_2 emissions, diesel emissions commence earlier. As illustrated and expected, diesel fuel emits less oxygen than ethanol/DME because of its chemical structure. Interesting to note, oxygen emissions asymptotes near zero as equivalence ratio borders unity for both fuels, this reinforces over-fuelling resulting in incomplete combustion beyond unity. This further explains the result of excessive THC produced for diesel fuel. As previously mentioned, ethanol and DME are oxygenated fuels, hence the higher concentration of specific emissions produced.

When comparing ethanol/DME 3 hole and 4 hole configurations, the 4 hole produces higher concentrations of oxygen until an approximate equivalence ratio of 0.6 is reached, beyond which the specific emissions are almost the same. This occurrence above $\phi \approx 0.6$ is unexpected, as the fuel conversion efficiency of the 3 hole is consistently higher than the 4 hole configuration, intuitively meaning improved combustion hence lower emissions of oxygen for the 3 hole configuration. However, chemical kinetics could have possibly produced this occurrence, as a higher concentration of oxygen is consumed to produce higher concentrations of CO and CO_2 for the 4 hole configuration.

Analysis after the catalytic converter

The discussion thus far only focused on the specific emissions produced before passing through the monolith of the catalytic converter. From the results depicted in the preceding graphs, the ethanol/DME 4 hole configuration, in most instances, produced the most favourable specific emissions. Due to this observation and in the interest of conciseness, the catalytic converter performance will only be viewed in terms of the ethanol/DME 4 hole specific emissions. Diesel fuel specific emissions after the catalytic converter will be used as a reference for the discussion of the ethanol/DME 4 hole specific emissions.

7.2.6 Specific NO_x emissions

As the exhaust gases pass through the monolith, rhodium, one of the metals found in the monolith prompts equation 7.2.4 below to occur.[1] This decreases NO_x by a reduction reaction using CO. The CO displaces and dissociates the NO since the CO bond possesses a higher bond energy than NO[28], and in this process liberates the oxygen molecule, producing CO_2 and N_2.

$$CO + NO \rightarrow CO_2 + \frac{1}{2}N_2 \quad [32] \qquad [7.2.4]$$

Figure 7.16 shows the specific emissions before and after the catalytic converter.

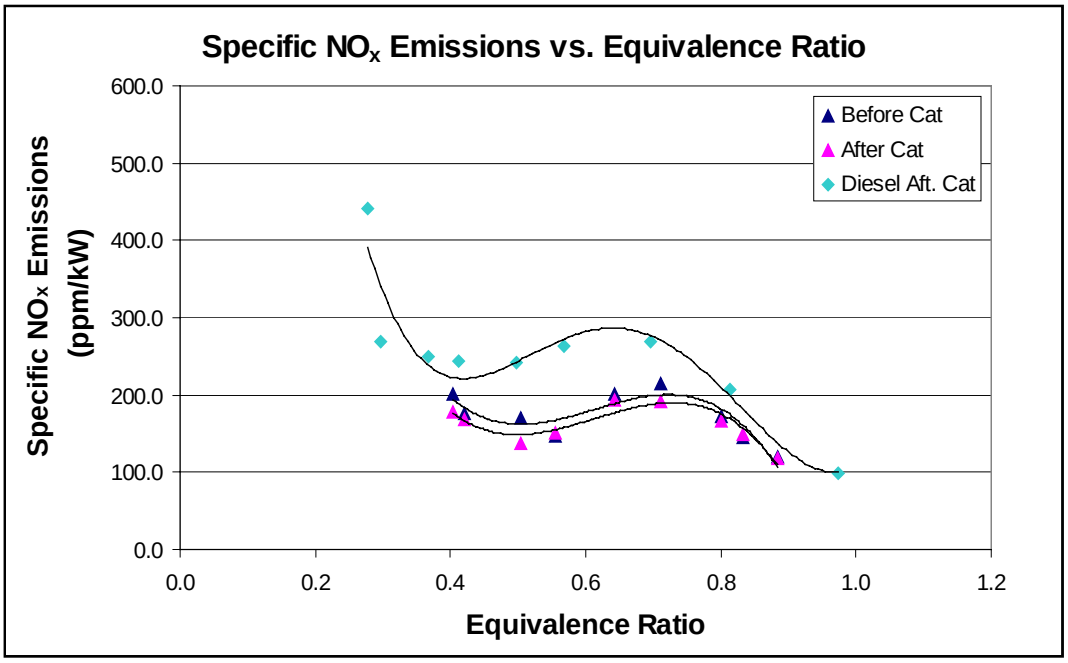

Figure 7.16: Specific NO_x emissions vs. Equivalence Ratio – Ethanol/DME 4 Hole

The largest reduction of NO_x for ethanol/DME 4 hole is achieved from $\phi \approx 0.40$ to $\phi \approx 0.59$. Referring to the discussion pertaining to figure 7.13, this is also the range at which CO emissions of ethanol/DME are considerably high before decreasing sharply. This fact emphasises equation 7.2.4 for the removal of NO. The NO molecules dissociate and the oxygen atoms which result, combine with CO to form CO_2. For $\phi \geq 0.59$, the difference before the catalytic converter and after the catalytic converter is very modest. According to Heywood[1], in order to remove NO effectively under lean conditions, the catalyst must react the CO, H_2 or HC with NO rather than O_2, as the exhaust gas passes through the catalyst bed. Ferguson[6] confirms this and has found that the removal of NO_x in diesel engines is very poor because the engine runs lean. Iwamoto et al[33] states the activity of the three-way catalyst for NO removal decreases to zero in the presence of just 0.5 vol % of O_2. Although the O_2 content in the gases as depicted in figure 7.15 are substantially higher than Iwamoto's findings, the CO emissions produced ensures that equation 7.2.4 occurs. It seems for $\phi \geq 0.59$, where CO emissions are low, Iwamoto's findings hold, where reactions with O_2 are dominant, hence the modest difference between emissions before the catalytic converter and after the catalytic converter shown.

Also worth noting, even though the removal of NO_x for ethanol/DME is poor, the specific emissions are still substantially lower than diesel after the catalytic converter emissions.

This may be contradictory to Heywood and Ferguson's observations, owing to the fact that diesel is a non-oxygenated fuel and ethanol/DME being oxygenated, expecting diesel emissions after the catalytic converter to be lower than ethanol/DME emissions. Due to the lower emissions produced prior to the gases passing through the catalytic converter, this overrides the oxygen content of the fuels.

7.2.7 Specific THC emissions

The THC emissions produced upstream of the catalytic converter undergo oxidation reactions when passing through the monolith. Sufficient oxygen from the exhaust stroke must be present to oxidise THC to CO_2 and water. The oxidation of THC is divided into two mechanisms, i.e. rapidly oxidised hydrocarbon, C_3H_6 representing 86% THC and slow oxidised hydrocarbon, CH_4 representing 14% THC[32]. The oxidation reactions representing these two mechanisms are respectively as follows:

$$C_3H_6 + \frac{9}{2}O_2 \rightarrow 3CO_2 + 3H_2O \quad [32] \qquad [7.2.5]$$

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \quad [32] \qquad [7.2.6]$$

Figure 7.17 shows the specific emissions before and after the catalytic converter.

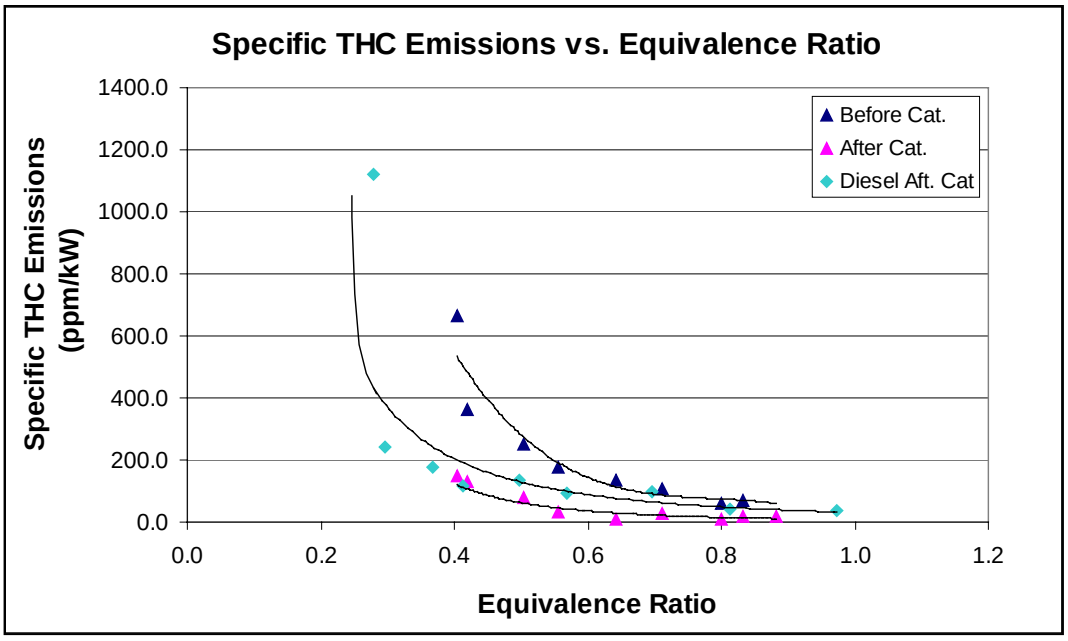

Figure 7.17: Specific THC emissions vs. Equivalence Ratio – Ethanol/DME 4 Hole

As depicted, the specific emissions after the catalytic converter is substantially lower than before the catalytic converter, amounting to almost 70% removal of THC. However, as the equivalence ratio increases, the gradients of the curves are different, with the after catalytic converter curves being less sloped. This difference illustrates the dependency on oxygen for the removal of THC – via equations 7.2.5 and 7.2.6. As the fuel becomes more rich, the local oxygen available becomes insufficient to support the oxidation reactions, even though conversion rates increase with increasing temperature.[34] When comparing the diesel specific emissions, they are lower than ethanol/DME specific emissions before the catalytic converter but higher than ethanol/DME specific emissions after the catalytic converter. The difference between diesel and ethanol/DME after the catalytic converter is attributed to the oxygen content of ethanol/DME fuel. A possible drawback to the oxygen content could be the increased formation of aldehydes within the catalytic converter. Aldehydes are formed by the incomplete oxidation of hydrocarbons and have been found to be carcinogenic and possess ozone formation potential. McCabe et al[35] discovered that platinum and palladium in the catalytic converter produced an increase in aldehyde concentration and that palladium catalyses the conversion from methanol (an alcohol much similar to ethanol) to formaldehyde (an aldehyde type) to a high extent. Thus, even though ethanol/DME emissions are lower than diesel emissions after the catalytic converter, aldehyde emissions could possibly be higher when analysing THC emissions in isolation.

7.2.8 Specific CO emissions

The last noxious exhaust gas, emanating from diesel engines to be discussed is CO. Similar to THC, CO also undergoes oxidation reactions when passing through the monolith of the catalytic converter and also requires sufficient oxygen from the exhaust stroke to oxidise CO into CO_2. The oxidation reaction is as follows:

$$CO + \frac{1}{2}O_2 \rightarrow CO_2 \quad [32] \qquad [7.2.7]$$

Figure 7.18 shows the specific emissions before and after the catalytic converter.

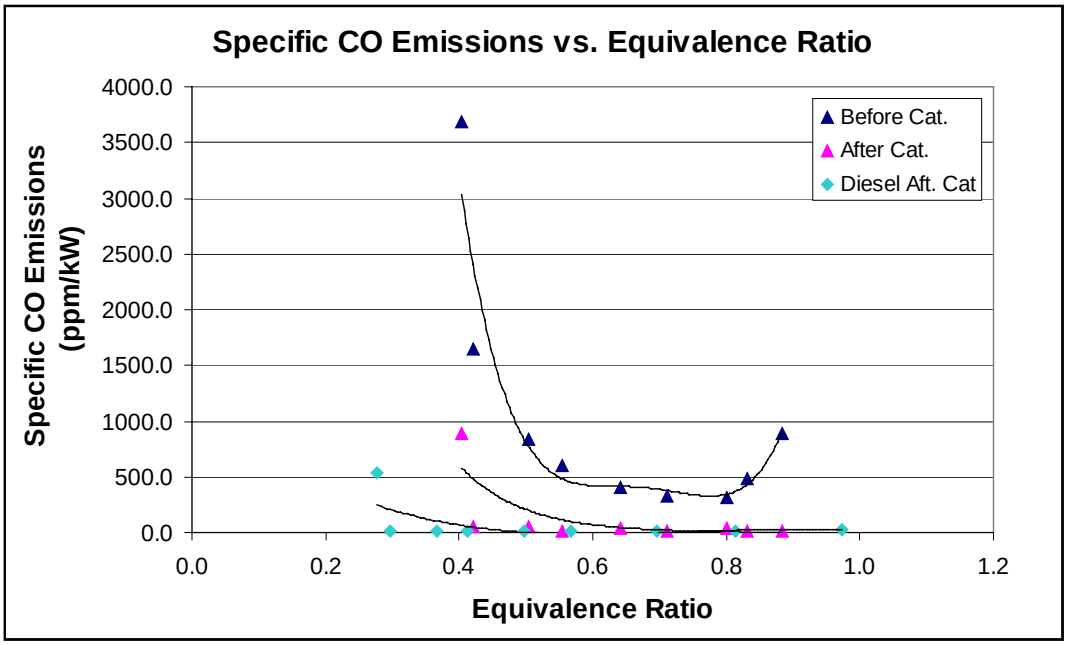

Figure 7.18: Specific CO emissions vs. Equivalence Ratio – Ethanol/DME 4 Hole

The after catalytic converter emissions are virtually zero. Besides the oxidation of CO molecules, the water which results from the oxidation of THC, as illustrated in equations 7.2.5 and 7.2.6, also plays a significant role in that it is consumed in the water-gas shift reaction, given as follows:

$$CO + H_2O \rightarrow H_2 + CO_2 \quad [1] \qquad [7.2.8]$$

When the water-gas shift equilibrium is approached, considerable CO is removed[1]. This is the probable reason for the trend shown in figure 7.18. This is especially the case for $\phi \geq 0.5$, where the concentration of H_2O molecules becomes more readily available to react with CO. It seems as ϕ increases, the oxidation mechanism transforms and is dominated by the water-gas shift mechanism, even though oxidation reactions are supported by temperature increases[31], reason being as ϕ increases, the catalytic converter warms up.

Catalytic converter efficiency with temperature variation will be covered later in the discussion.

7.2.9 Specific CO_2 emissions

The production of CO_2 by internal combustion engines is not a major contributor to the concentration of greenhouse gas found in the atmosphere, hence not considered a noxious exhaust gas.[2] The preceding equations, 7.2.4 to 7.2.8, reduction-oxidation mechanisms produced CO_2 as a product after reactions. Figure 7.19 shows the specific emissions before and after the catalytic converter.

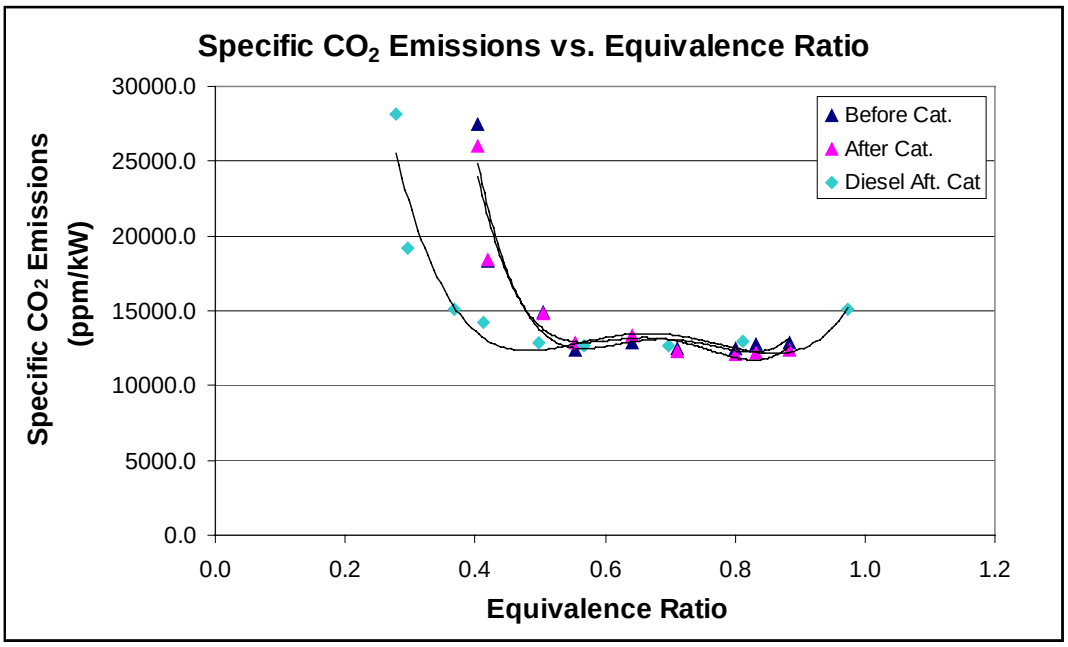

Figure 7.19: Specific CO_2 emissions vs. Equivalence Ratio – Ethanol/DME 4 Hole

Immediately evident in the trend depicted in figure 7.19 is the zero difference between ethanol/DME before the catalytic converter and after the catalytic converter emissions. This is unexpected as more CO_2 should be produced from equations 7.2.4 to 7.2.8. It is speculated that this could be due to the failure of recombination reactions with the oxygen and carbon molecules succeeding the reduction-oxidation reactions. However, confirmation of this will only be established next, when the specific O_2 emissions are discussed. Diesel specific emissions commence at $\phi \approx 0.23$ as compared to ethanol/DME's delayed commencement at $\phi \approx 0.41$, largely dictated by the mass flowrate of emissions before the catalytic converter.

The trend of the two fuels are the same for $\phi \geq 0.54$, however the ethanol/DME trend curves are expected to be above the diesel fuel trend curve, because of the difference in the chemical structure between the two fuels.

7.2.10 Specific O_2 emissions

The oxygen molecules that arise in the catalytic converter are used for the oxidation reactions occurring in THC and CO, previously described. Figure 7.20 shows the specific emissions before and after the catalytic converter.

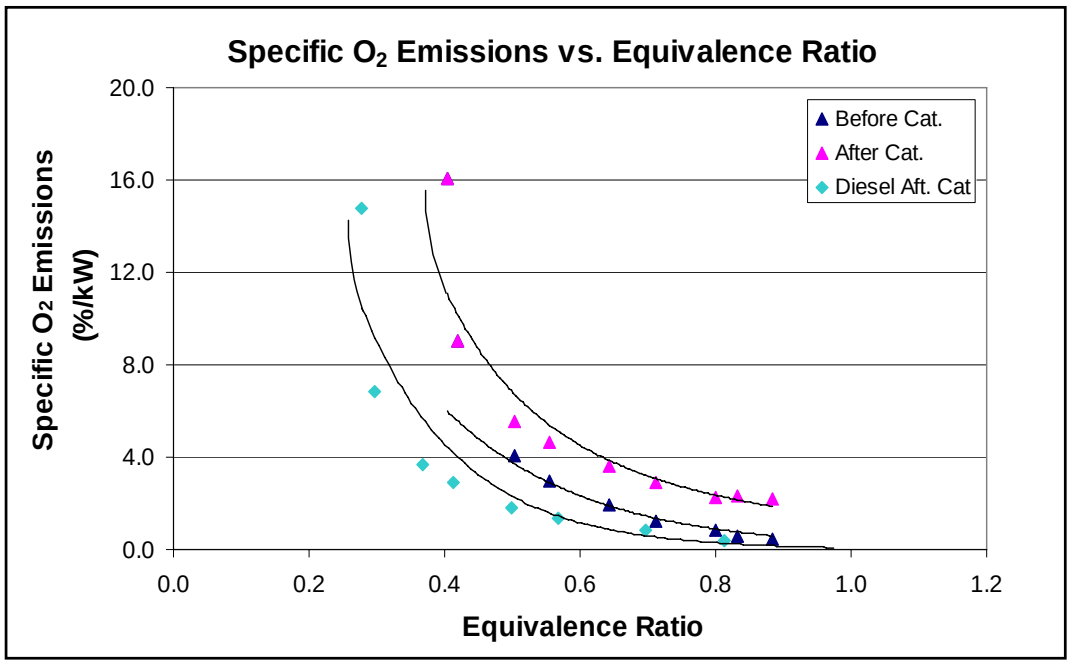

Figure 7.20: Specific O_2 emissions vs. Equivalence Ratio – Ethanol/DME 4 Hole

As depicted, the specific emissions after the catalytic converter are higher than the specific emissions before the catalytic converter. This result removes the speculation raised earlier with regard to the CO_2 emissions after the catalytic converter. It seems that recombination reactions of the oxygen molecules with the carbon molecules do not occur. Further to these recombination reactions, questions may arise as to whether the dissociation reactions for THC and CO actually do occur, when undergoing initial oxidation within the monolith. The answer lies in the removal of THC and CO, as depicted in figures 7.17 and 7.18 respectively, which proves that dissociation does indeed occur. However, the oxygen molecules liberated by dissociation escape and fail to recombine with the carbon molecules to form CO_2.

In addition, the formation of water droplets at the tail end of the exhaust hose was also observed. It could be that a portion of the oxygen molecules reacted with hydrogen, found in the THC dissociation process, to form water.

The oxidation reaction representing this formation is as follows:

$$H_2 + \frac{1}{2}O_2 \rightarrow H_2O \quad [32] \qquad [7.2.9]$$

Lastly, when comparing diesel with ethanol/DME, diesel specific emissions after the catalytic converter are even lower than ethanol/DME specific emissions before the catalytic converter, once again attributed to the oxygen content of the ethanol/DME fuel.

7.2.11 Specific Smoke emissions

Diesel particulates or smoke forms as a result of the incomplete combustion process of hydrocarbons, and consists primarily of carbonaceous material on which some organic compounds have become absorbed.[1] According to current theories, the carbon particles can be formed from hydrocarbon fuels in the presence or absence of oxygen.[8] The reaction shown illustrates the path for the formation of these carbon particles.

$$C_\alpha H_\beta + \frac{\chi}{4}O_2 \rightarrow \frac{\chi}{2}CO + (\alpha - \frac{\chi}{2})C(s) + \frac{\beta}{2}H_2 \quad [6] \qquad [7.2.10]$$

When there is insufficient oxygen present to convert all the carbon in the fuel to carbon monoxide, for $2\alpha > \chi$, then solid carbon or smoke is produced. This is likely to occur locally within the fuel spray injected into the combustion chamber, since it takes time for the oxygen to be mixed with the fuel.[6] Figure 7.21 shows the variation of specific smoke emissions for increasing equivalence ratio.

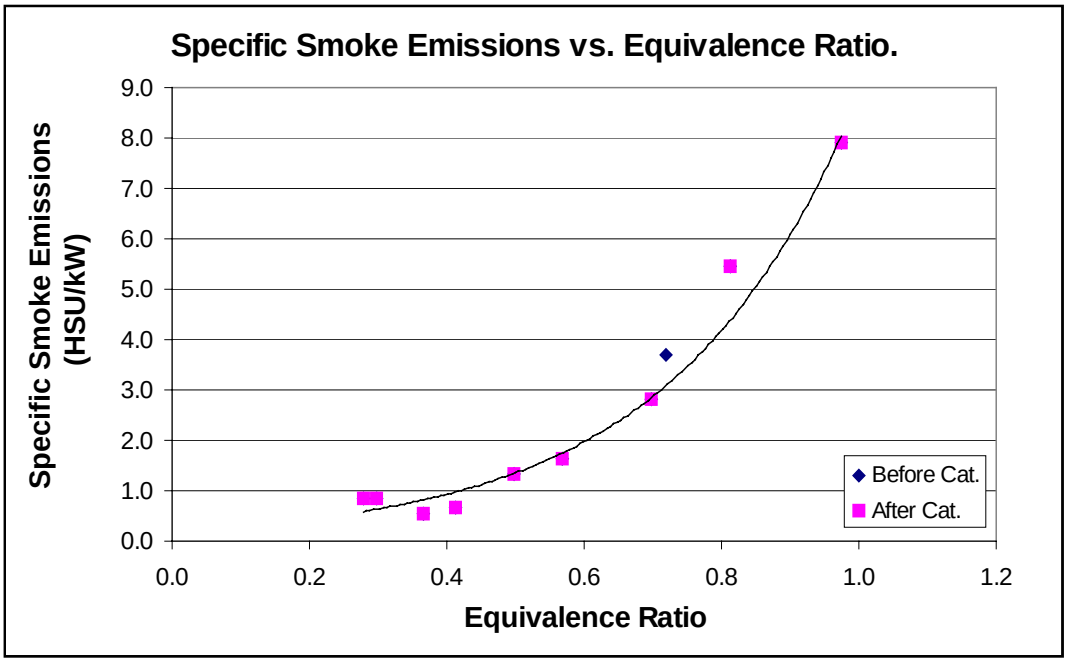

Figure 7.21: Specific Smoke emissions vs. Equivalence Ratio – Diesel

Ethanol/DME smoke emissions are insignificant when compared to diesel smoke emissions, hence are not shown in figure 7.21. The diesel specific smoke emissions depicted increase in a parabolic fashion, for increasing equivalence ratio. This trend is as expected, since more unburned hydrocarbons result, and less oxygen becomes available for near complete combustion, as ϕ approaches unity. This is especially true for high loading resulting in maximum power output being limited by smoke emissions. Ethanol/DME's almost zero smoke emissions is due to its oxygen content and by virtue of this, $\chi \geq 2\alpha$ in equation 7.2.10 is always maintained. Previous work conducted[36], showed the oxygen content of a fuel provided an improvement in fuel evaporation, and diffusion combustion was more pronounced at higher bmep. Ethanol/DME therefore has the potential to increase rated power without incurring the penalty of increased smoke emissions, as would be the case for diesel fuel.[37]

When comparing specific smoke emissions before and after the catalytic converter, as shown in figure 7.21, there exists no difference, and the reduction-oxidation mechanisms of the catalytic converter has no effect on smoke. Springer[8] has also found that catalytic converters have minimal effect on smoke emissions. The reason for this is the difficulty in oxidising solid carbon particles in the exhaust gas.

By the time the particles reach the monolith, cooling of the exhaust gas occurs, which in turn cools the carbon particles, further solidifying the particles. The water which forms within the monolith, increases the vapour content which further adds to the cooling of the carbon particles.

The converse of this would actually decrease specific smoke emissions after the catalytic converter, because the regenerative oxidation reactions of carbon particles are endothermic in nature.[1]

7.2.12 NO_x - Smoke Trade off

A major drawback with diesel fuel is the inability to reduce NO_x and smoke emissions simultaneously. NO formation predominantly occurs in the fuel-lean zones of the spray, during the premixed phase, whereas smoke is predominantly formed in fuel-rich zones of the spray, during the diffusion burning period of the combustion cycle. Due to these formation mechanisms, an inverse relationship exists between NO_x and smoke emissions. Figure 7.22 below shows the variation of NO_x with smoke for increasing load.

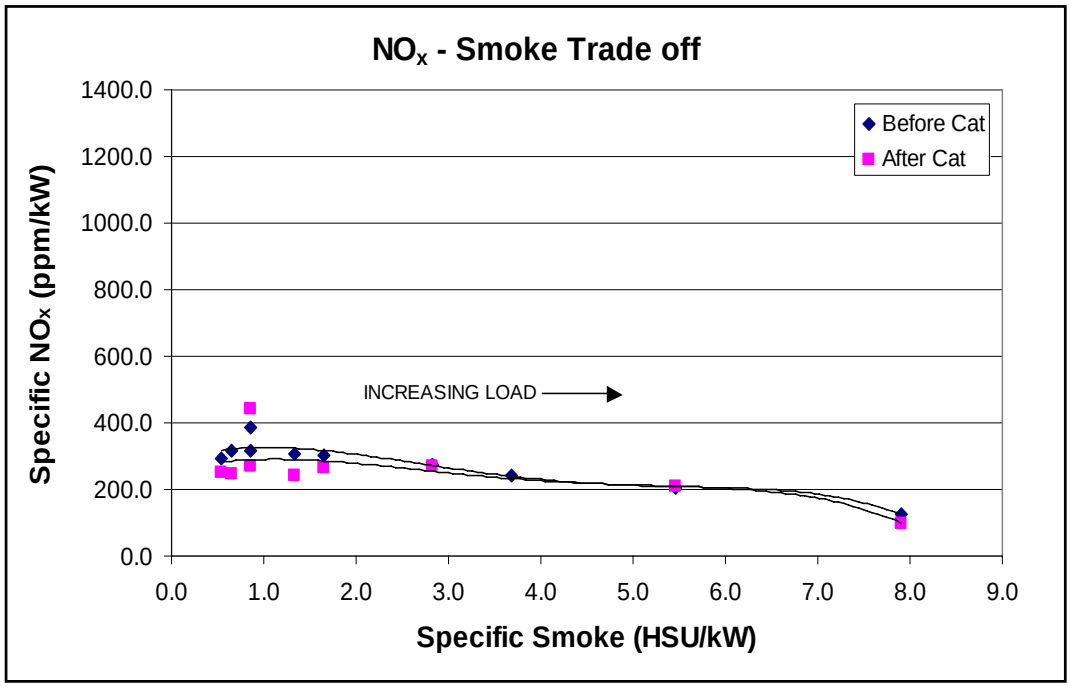

Figure 7.22: NO_x – Smoke Trade off – Diesel

Immediately evident in figure 7.22, is the decrease in specific NO_x emissions, while the specific smoke emissions increase. The specific smoke emissions have been plotted as the independent variable because maximum power output is limited by smoke emissions as compared to NO_x emissions, which possesses less influence on maximum power output.

The specific emissions before the catalytic converter are very similar to the specific emissions after the catalytic converter, mainly due to the difference existing for the specific NO_x emissions. As mentioned earlier in 7.2.1, NO_x formation is highly temperature dependent, and during premixed combustion, prevailing high temperatures facilitate and promote NO_x formation.

Efforts to reduce premixed combustion to contain NO_x formation have the opposite effect on smoke[38], when the premixed combustion phase is decreased, the diffusion burning period is increased. The latter part of diffusion burning, with reduced in-cylinder temperature and lesser oxygen available for complete combustion, has the tendency to produce more smoke. An advantage to note, the NO_x-smoke trade off occurrence does not exist for ethanol/DME fuel due to the smoke emissions being virtually zero for ethanol/DME, hence solely presenting the challenge of reducing NO_x emissions for increasing load in diesel engines.

7.3 Catalytic Converter Efficiency

The performance of the catalytic converter can only be measured by its ability to remove noxious emissions from the exhaust gas. The manner in which this is quantified is by the calculation of the catalytic converter efficiency, as described in the literature survey using equation 3.3.2[1]. The property which affects the performance of the catalytic converter is its temperature, since the activation of the reduction-oxidation reactions are dependent on the temperatures reached within the catalytic converter. Figures 7.23 to 7.25, show the temperature variation upstream and downstream of the catalytic converter for increasing equivalence ratio.

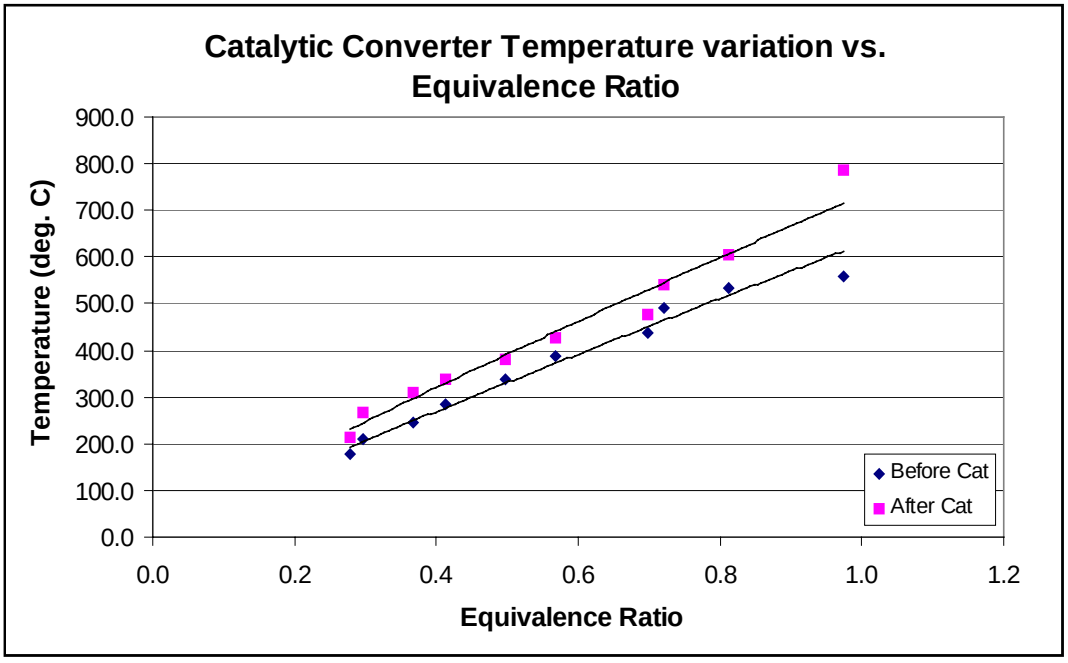

Figure 7.23: Temperature variation vs. Equivalence Ratio - Diesel

[1] Refer to Appendix F for a sample calculation.

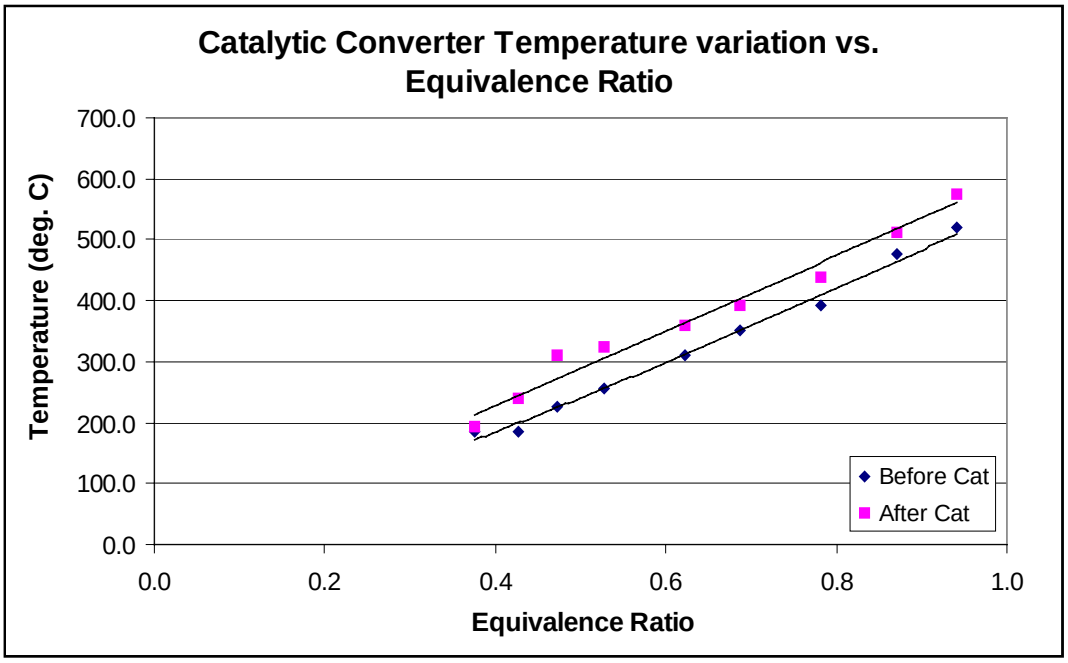

Figure 7.24: Temperature variation vs. Equivalence Ratio – Ethanol/DME 3 Hole

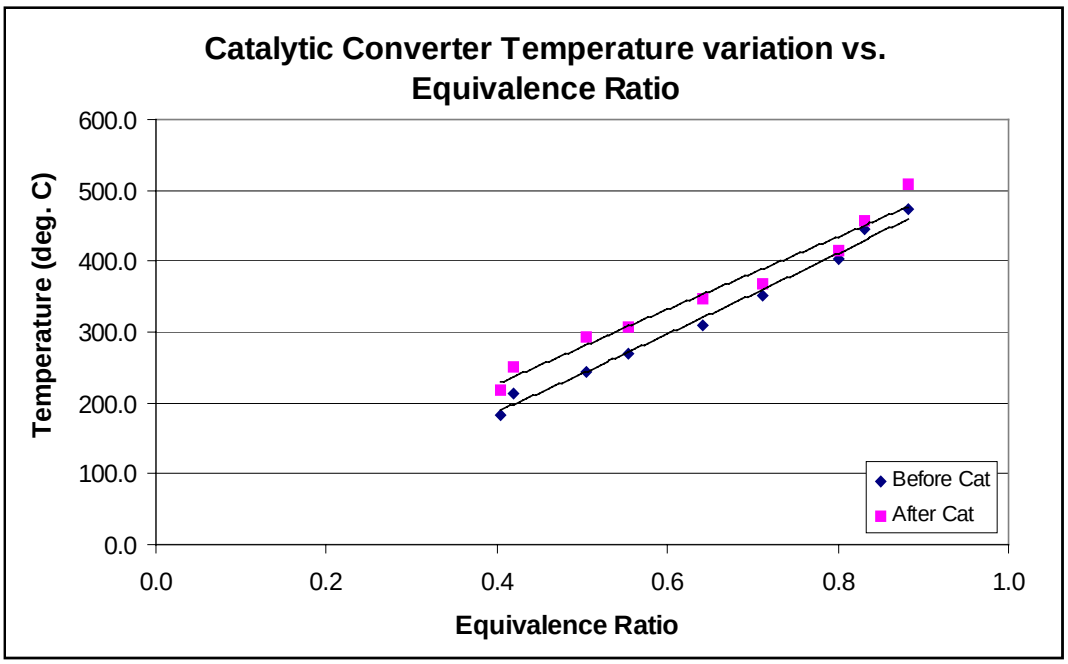

Figure 7.25: Temperature variation vs. Equivalence Ratio – Ethanol/DME 4 Hole

The temperature increases for increasing equivalence ratio. The reduction-oxidation reactions which occur in the monolith are exothermic in nature[39], hence the energy released and the convective heat transfer result in an increase in temperature as the mass transfer of exhaust gas increases. Furthermore, from figures 7.23 to 7.25, the temperatures after the catalytic converter are higher than the temperatures before the catalytic converter, also due to the energy release of the reduction-oxidation reactions. However, when comparing the differential between the temperatures before the catalytic converter and after the catalytic converter of the fuels, diesel fuel possesses the greatest difference, followed by ethanol/DME 3 hole and lastly ethanol/DME 4 hole. This means the diesel fuel reduction-oxidation reactions release the most energy. The reason for diesel possessing this greatest difference is due to the amount of energy released during the oxidation of THC, as well as the chemical composition of hydrogen and carbon found in diesel fuel.[39]

When analysing the efficiencies obtained for the noxious emissions, the temperature after the catalytic converter is used, because the assumption that catalytic activity occurs throughout the length of the monolith forms the basis of the discussion. In conjunction, the light-off temperature is especially important, because this is the temperature where the catalytic converter becomes 50% efficient for a particular emission species. Figure 7.26 shows the catalytic converter efficiency for NO_x emissions.

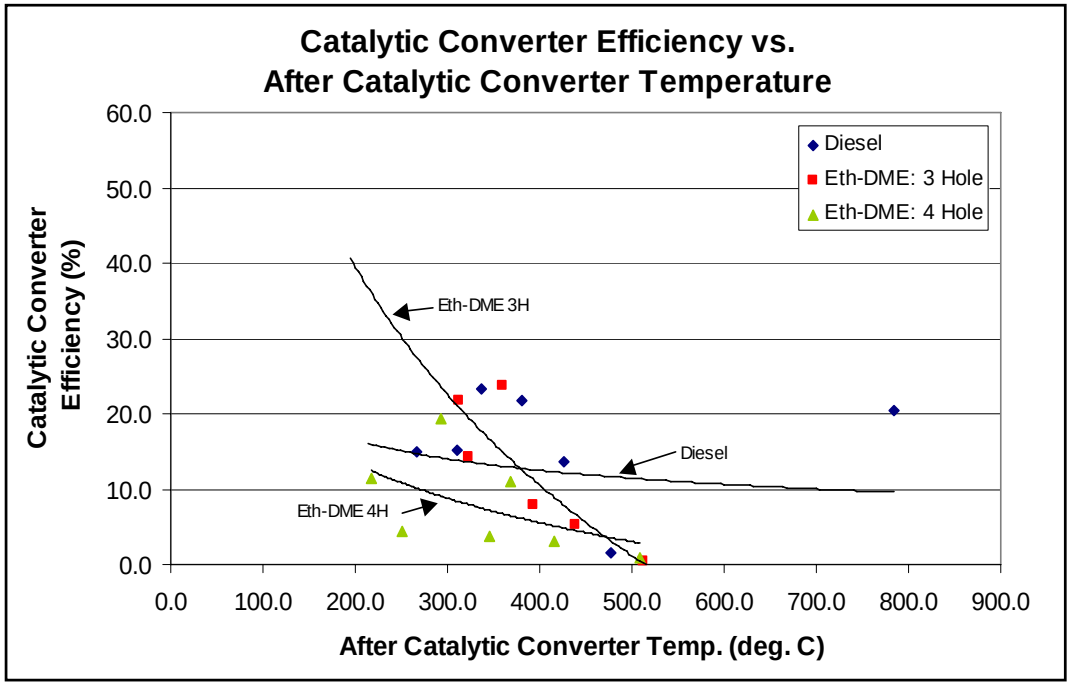

Figure 7.26: Catalytic Converter Efficiency vs. After Catalytic Converter Temperature for NO_x emissions

As depicted, ethanol/DME 3 hole is the most efficient up until a temperature of approximately 380^{o}C, afterwhich diesel is more efficient. The efficiencies for both ethanol/DME and diesel are very low, to the extent that light-off temperatures are never reached. As mentioned previously, this is mainly due to the lean combustion of diesel engines. However, substantial improvements exist in the removal of THC, as shown in figure 7.27.

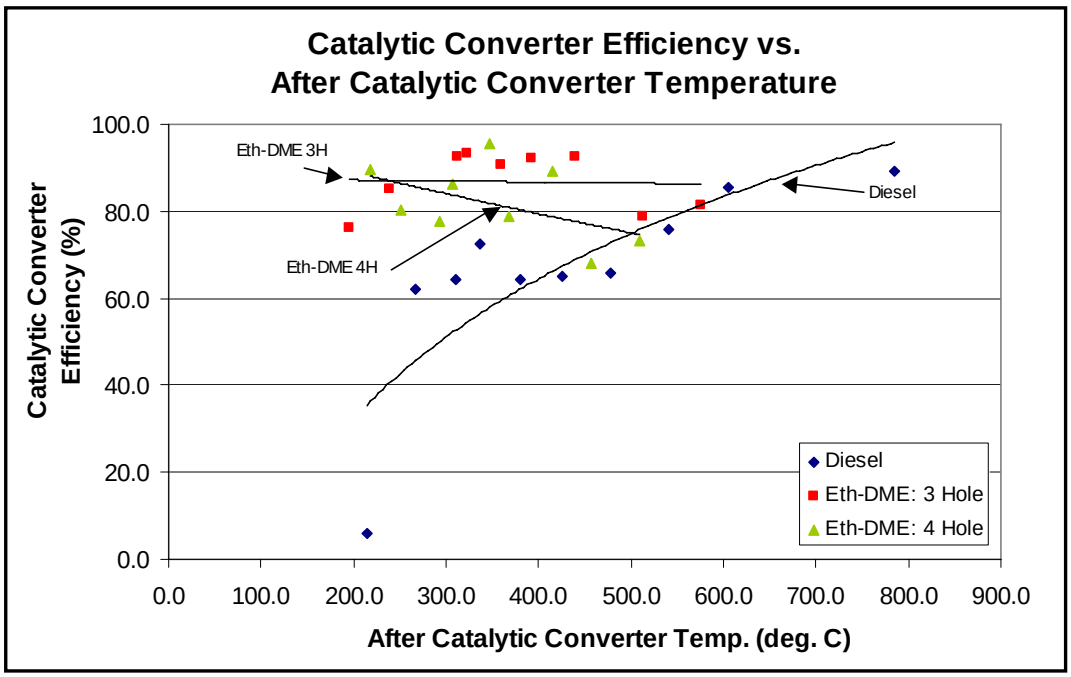

Figure 7.27: Catalytic Converter Efficiency vs. After Catalytic Converter Temperature for THC emissions

Ethanol/DME 3 hole efficiencies are between 80% and 90%. Ethanol/DME 4 hole efficiency begins close to 90% then tapers off to approximately 70%. Interesting to note, for both the 3 hole and the 4 hole ethanol/DME efficiencies, there exists no light-off temperature because the efficiencies reached are well above 50%. However, from the trend curve diesel fuel does possess a light-off temperature of approximately 290^{o}C. The first diesel efficiency point shown in figure 7.27 seems unusually low at approximately 6%, but this is possible and mainly attributed to the catalytic converter not reaching its operating temperatures. Diesel efficiencies then steadily increase to a maximum of approximately 90% as the catalytic converter warms up.

When analysing the efficiencies for CO emissions as shown in figure 7.28, similar trends exist as in THC.

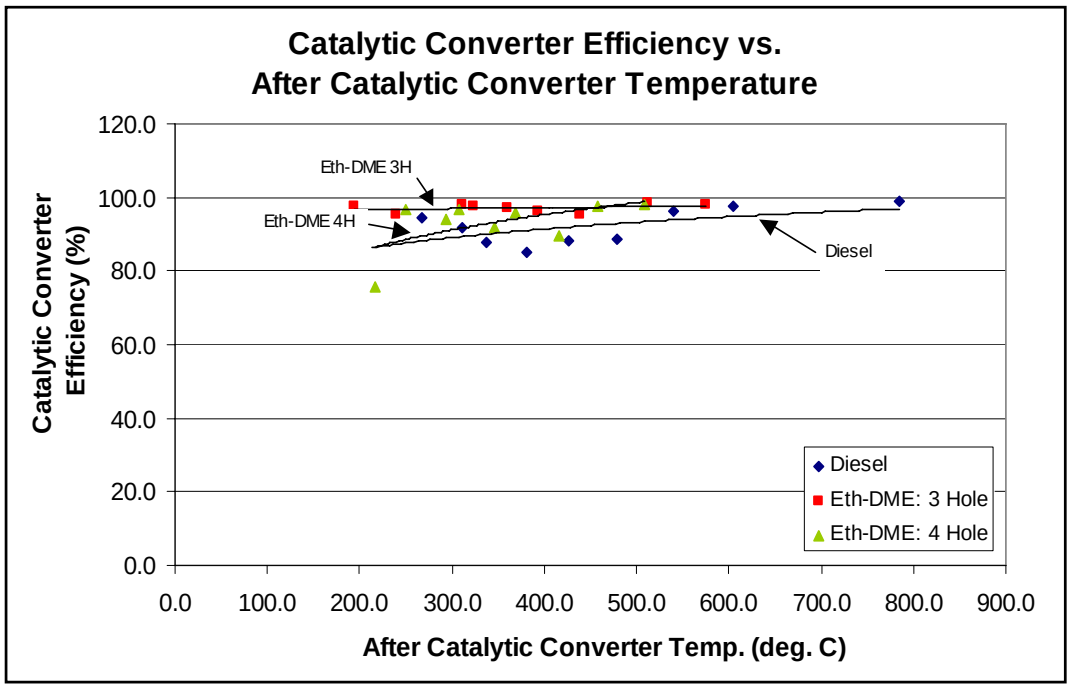

Figure 7.28: Catalytic Converter Efficiency vs. After Catalytic Converter Temperature for CO emissions

Ethanol/DME 3 hole efficiencies hovers around 98%, whereas ethanol/DME 4 hole and diesel efficiencies increase steadily. From the preceding figures, evidence shows that the catalytic converter removes THC and CO more readily. The efficiencies of ethanol/DME are particularly high from the onset, showing that the oxidation mechanism for ethanol/DME is not highly temperature dependent to the same extent diesel is. Thus, conclusion can be made, that oxygen content in the fuel plays a substantial role in the performance characteristics of the catalytic converter.

8 CONCLUSION

The preceding chapters leading upto the discussion of results, have shown the viability of ethanol as an alternative fuel in conventional compression ignition engines. DME was used as an ignition promoter when the engine was fuelled on ethanol. Although the engine was optimised for diesel fuel combustion, majority of the trends observed were as expected, with the assistance of literature from previous work conducted on alcohol fuels at the University and other authors. However, unique to this research was the use of a catalytic converter in diesel exhaust emissions analysis. This application has provided further reduction in emissions, thereby reinforcing ethanol's attractiveness as an alternative fuel. The main findings described below pertain to the performance, emissions and catalytic converter efficiency.

8.1 Performance

- The brake power output of ethanol/DME fuel was comparable to diesel fuel, even though more fuel was required in the ethanol/DME case. For diesel, testing was limited by excessive black smoke being emitted, whereas for ethanol/DME this limitation did not exist. Without compromising other performance parameters, the possibility of trying to increase the brake power output of the engine when fuelled with ethanol/DME as such does exist.

- The brake specific fuel consumption for ethanol/DME was higher than diesel. This trend was no surprise as ethanol and DME possess lower calorific values than diesel, meaning the energy content within diesel is higher. For both fuels the specific fuel consumption was notably high at low loads, as optimum operating conditions were not reached in the combustion chamber.

- The specific fuel conversion efficiency of diesel was higher and occurred at lower equivalence ratios than ethanol/DME. The differences between the nozzle configurations in ethanol/DME were slightly more pronounced than the two former performance parameters mentioned, with the ethanol/DME 3 hole nozzle consistently presenting higher efficiencies than ethanol/DME 4 hole.

These differences were largely driven by the energy release characteristics of the fuels and nozzle configurations. The pressure traces revealed a close correlation between the point of energy release before TDC and the fuel specific conversion efficiency attained. Energy released closer to TDC results in less negative work being done by the piston, thereby presenting higher fuel conversion efficiencies.

- The ignition delay for both the fuels decreased with increasing brake power. The ignition delay of ethanol/DME was however lower than that of diesel fuel. This occurrence was unexpected as the ignition quality of ethanol is poorer than diesel. However, this did show the ignition enhancing effect that DME possessed.

- The combustion chamber pressure and temperature increased for increasing equivalence ratio. The difference between diesel and ethanol/DME was very modest. From fundamental physics principles, when pressure increases, temperature increases and in the case of diesel fuel slightly higher pressures were attained, hence slightly higher combustion temperatures reached. Although ethanol/DME pressures and temperatures were slightly lower than diesel, it was largely higher than initially expected. This is due to DME's two stage energy release pattern, in which the overall energy release predominantly consists of the second stage release, as it increases with increasing brake power.

- The mechanical efficiency of ethanol/DME 4 hole was very close to that of diesel, especially near stoichiometry. Ethanol/DME 3 hole mechanical efficiencies were exceptionally low throughout the load range. This may seem ironic since the 3 hole consistently produced superior performance results for the parameters mentioned above, compared to the 4 hole. Consequently, frictional losses in ethanol/DME 3 hole was greater.

8.2 Emissions

- Specific NO_x emissions before the catalytic converter of diesel fuel was higher than ethanol/DME at low loads. Ethanol/DME specific emissions deteriorated as equivalence ratio increased, however they were still lower than diesel. Ethanol/DME 4 hole produced the least NO_x emissions before the catalytic converter. The specific emissions after the catalytic converter were very similar to the specific emissions before the catalytic converter for ethanol/DME 4 hole. These results showed the non-linear temperature effects on NO_x emissions.

- Specific THC emissions before the catalytic converter of the two fuels initially decreased until mid-load ranges were reached. Thereafter, opposing trends existed, with ethanol/DME specific THC emissions decreasing while diesel specific THC emissions increased. This was mainly due to the oxygen content in ethanol/DME. Ethanol/DME 4 hole produced the least specific THC emissions before the catalytic converter, as well as after the catalytic converter. However, diesel THC results were comparable to ethanol/DME 4 hole after the catalytic converter.

- Specific CO emissions before the catalytic converter produced by diesel and ethanol/DME possessed the most profound opposing trends at low and high loads, when compared to THC specific emissions. This occurrence was mainly due to chemical kinetics in the combustion chamber. The ethanol/DME 3 hole and 4 hole difference was very modest. Diesel and ethanol/DME 4 hole specific emissions after the catalytic converter were virtually zero throughout the load range.

- Specific CO_2 emissions before and after the catalytic converter were very similar. However in the case of ethanol/DME 4 hole, specific emissions after the catalytic converter were expected to be higher. It was assumed that the failure of recombination reactions in the reduction-oxidation mechanisms of the catalytic converter caused this and thus still requires further verification.

- Specific O_2 emissions decreased for increasing equivalence ratio. In both, before and after the catalytic converter, ethanol/DME produced more oxygen emissions. This was as expected because of the higher oxygen content found in ethanol/DME.

- Specific smoke emissions increased exponentially for increasing equivalence ratio. This was only applicable to diesel fuel. Ethanol/DME's specific smoke emissions were negligible when compared to diesel, as a result NO_x-smoke trade offs commonly encountered in compression ignition engines do not pose as a limiting factor in ethanol/DME fuelling. Also for diesel fuel, the variation in NO_x emissions was substantially less than smoke emissions for increasing load. It seems the catalytic converter has no effect on smoke emissions.

8.3 Catalytic Converter Performance

- The temperature within the catalytic converter increased with increasing equivalence ratio. As the temperature increased the performance of the catalytic converter improved. The temperature after the catalytic converter was found to be slightly higher than before the catalytic converter. This variation was greater in diesel than with ethanol/DME fuelling.

- The reduction-oxidation reactions, which occur in the monolith of the catalytic converter are exothermic in nature and the energy release characteristics cause the temperature rise in the catalytic converter.

- Of the three noxious gases analysed, the catalytic converter is most efficient in removing CO, followed by THC and lastly NO_x emissions, which shows the tendency to favour oxidation reaction mechanisms as compared to reduction reaction mechanisms.

- Ethanol/DME 3 hole is the most efficient for the removal of the three noxious gases, followed by ethanol/DME 4 hole and lastly diesel fuel.

- None of the fuels reached light-off temperature for NO_x emissions, whereas for CO emissions, light-off temperatures are non-existent since the efficiencies are above 50% from the very onset. For THC emissions only diesel reached light-off temperature whereas ethanol/DME efficiencies are above 50% from the very onset.

- Overall, the ethanol/DME catalytic converter efficiency characteristics are superior to those of diesel fuel, mainly due to the oxygen contained in ethanol/DME's chemical composition. This shows that in order for the catalytic converter to yield best results, oxygen must be present.

8.4 Summary

With the research tests conducted and subsequent conclusions drawn, ethanol has shown to be a distinct contender in replacing diesel as combustible fuel in compression ignition engines. Ethanol's combustion and emissions characteristics were largely due to its physical and chemical properties. Ethanol/DME has proved that in terms of performance of the engine, it nearly matches diesel's performance characteristics and with further optimisation methods, most certainly has the potential to surpass diesel's performance characteristics.

By far the most significant results obtained from this research was the reduction in the emissions, mainly due to ethanol and DME's oxygen content, producing far more superior and favourable emissions than diesel. In addition to this, incorporating a catalytic converter to compression ignition emission analysis proved to be very successful, providing the scope for further in-depth analysis in future research topics.

The only limiting factor facing ethanol as an alternative fuel, is the relatively high cost associated with its production. The raw material resource availability for production of ethanol is no problem. However, with mounting tensions being experienced worldwide arising from geo-political hostile takeovers for possession of limited and depleting crude oil resources, production costs of ethanol would seem insignificant. Another drawback, but of lesser degree is the lower volumetric energy density of ethanol, which means future ethanol vehicles would require larger fuel tanks for acceptable coverage.

Presently, the environmental advantages linked to ethanol combustion, has the potential to outweigh the production cost implications. With government institutions and regulatory organisations assistance, wide scale distribution and mass production fuelling infrastructure could eleviate production cost concerns and provide market awareness, and introduction of ethanol as a fuel of the future.

9 RECOMMENDATIONS

The research work conducted in this report is merely a part of a vast topic of ethanol combustion and emissions. In the near future demand for research of this nature will become more prevalent as the need for cleaner air increases. Several recommendations to assist future researchers in this field are listed below.

- Optimising the compression ignition engine parameters to best suit ethanol/DME combustion, thereby improving brake power output and fuel conversion efficiency.

- Possibly increasing injector pressure and retarding injection timing to improve energy release patterns, resulting in minimal negative work being done by the piston. This will improve performance characteristics of ethanol/DME fuel.

- Since the 4 hole nozzle configuration using ethanol/DME fuel yielded the best results at an engine speed of 1550 rpm, other speeds should also be investigated to establish if the same observations hold.

- Increasing the energy content of ethanol by adding higher calorific value additives or emulsions of diesel to reduce brake specific fuel consumption.

- Introducing exhaust gas recirculation into the emissions analysis. EGR has been known to further reduce NO_x emissions in diesel engines, especially at higher loads.

- Analysing another oxygenated fuel such as DMC, and comparing it with methanol and ethanol fuels. Also possibly investigating neat DME combustion with EGR.

- Possibly investigating a more controlled manner in which to aspirate DME during the air intake phase of combustion to further enhance ignition of ethanol.

- Conducting heat transfer research using ethanol/DME and comparing it with diesel fuel and compiling data for computational thermodynamic software simulation packages, this having the advantage of performing analysis without laboratory research work.

- Finally, laboratory research work has some limitations, the results seem promising in the laboratory environment, however real time city driving will be the ultimate test. Therefore, diesel automobiles should be fuelled with ethanol to observe real time performance results.

10 REFERENCES

1. Heywood, J. B. *Internal Combustion Engine Fundamentals, 1st International Edition*, McGraw-Hill, Inc., 1988.

2. Plint, M. and Martyr, A. *Engine Testing Theory and Practice, 2nd Edition*, Butterworth-Heinemann, 1999.

3. Milton, B. E. *Thermodynamics, Combustion and Engines, 1st Edition*, Chapman and Hall, 1995.

4. Wang, H. W., Zhou, L. B., Jiang, D. M. and Huang, Z. H. *Study on the performance and emissions of a compression ignition engine fuelled with Dimethyl ether*, Proceedings of the Institute of Mechanical Engineers, Volume 214, Part D, 2000, pp 101 – 106.

5. Ecklund, E. E., Bechtold, R. L., Timbario, T. J. and McCallum, P. W. *State of the Art Report on the Use of Alcohols in Diesel Engines*, SAE Paper 840118, 1984.

6. Ferguson, C. R. and Kirkpatrick, A. T. *Internal Combustion Engines, Applied Thermosciences, 2nd Edition*, John Wiley and Sons, Inc., 2001.

7. Borman, G. L. and Ragland, K. W. *Combustion Engineering, 1st International Edition*, McGraw-Hill Book Co., 1998.

8. Springer, G. S. and Patterson, D. J. *Engine Emissions – Pollutant Formation and Measurement, 1st Edition*, Plenum Press NY, 1973.

9. Scott, W. M. *Engine Emissions – Pollutant Formation and Measurement*, Plenum Press NY, 1973, Chapter 6, number 29, pp 262.

10. Henein, N. A. *Engine Emissions – Pollutant Formation and Measurement*, Plenum Press NY, 1973, Chapter 6, number 8, pp 262.

11. El Wakil, M. M. *Engine Emissions – Pollutant Formation and Measurement*, Plenum Press NY, 1973, Chapter 6, number 24, pp 262.

12. Glikin, P. E. *Fuel Injection in Diesel Engines*, Proceedings of the Institute of Mechanical Engineers, Volume 199, Part D, 1985, pp 161 – 174.

13. Greeves, G. *Response of Diesel combustion systems to an increase in fuel injection rate*, SAE Paper 790037, 1979.

14. Khan, I. M., Greeves, G. and Wang, C. H. T. *Factors affecting smoke and gaseous emissions from direct injection engines and a method of calculation*, SAE Paper 730169, 1973.

15. Greeves, G. and Wang, C. H. T. *Origins of Diesel particulate mass emissions*, SAE Paper 810260, 1981.

16. Russell, M. F. and Cavanagh, C. J. *Establishing targets for control of Diesel Combustion noise*, SAE Paper 770259, 1977.

17. Russell, M. F. *Recent CAV Research into noise, emissions and fuel economy of diesel engines*, SAE Paper 770257, 1977.

18. Greeves, G. *Origins of Hydrocarbon emissions from Diesel engines*, SAE Paper 770259, 1977.

19. Cipolat, D. *Improvement in Performance of a Methanol Fuelled Compression Ignition Engine*, Proceedings of the XII International Symposium on Alcohol Fuels, Beijing China, 1998, pp 133 – 138.

20. Cipolat, D., Jawurek, H. H. and Rallis, C. J. *Methanol-DME fuelling of a compression ignition engine*, Proceedings of the IX International Symposium on Alcohol Fuels, Volume II, Firenze, 1991, pp 411 – 415.

21. Cipolat, D. *Emissions and combustion characteristics of a Methanol-DME dual-fuelled compression ignition engine*, Proceedings of the 3rd International Conference on Combustion Technologies for a Clean Environment, Section 9, Lisbon, 1995, pp 8 – 15.

22. Cipolat, D. *Emissions and performance characteristics of a compression ignition engine fuelled on methanol-DME*, Proceedings of the 4th International Conference on Combustion Technologies for a Clean Environment, Volume 1, Section 12, Lisbon, 1997, pp 27 – 33.

23. Pirouzpanah, V., Khoshbakhti Sarai, R. *Reduction of Emissions in an automotive Direct Injection Diesel engine dual-fuelled with Natural gas by using variable Exhaust Gas Recirculation*, Proceedings of the Institute of Mechanical Engineers, Volume 217, Part D, 2003, pp 714 – 725.

24. Huang, Z. H., Jiang, D. M., Zeng, K., Liu, B. and Yang, Z. L. *Combustion characteristics and Heat release analysis of a Direct injection compression ignition engine fuelled with Diesel-Dimethyl Carbonate blends*, Proceedings of the Institute of Mechanical Engineers, Volume 217, Part D, 2003, pp 595 – 605.

25. Huang, Z. H., Lu, H. B., Jiang, D. M., Zeng, K., Liu, B., Zhang, J. Q. and Wang, X. B. *Engine performance and emissions of a compression ignition engine operating on Diesel-Methanol blends*, Proceedings of the Institute of Mechanical Engineers, Volume 218, Part D, 2004, pp 435 – 447.

26. Kajitani, S., Chen, C. L., Oguma, M., Alam, M. and Rhee, K. T. *Direct Injection Diesel Engine operated with Propane – DME Blended Fuel*, SAE Paper 982536, 1998.

27. Cipolat, D. *Study of a Compression Ignition Engine fuelled on Ethanol with an ignition promoter*, Proceedings of the XIII International Symposium on Alcohol Fuels, Stockholm, 2000, pp 1 – 10.

28. Gillespie, R. J., Humphreys, D. A., Colin Baird, N. and Robinson, E. A. *Chemistry, 2nd Edition*, Prentice-Hall International, Inc., 1989.

29. Cheng, W., Hamrin, D., Heywood, J., Hochgreb, S., Min, K. and Norris, M. *An Overview of Hydrocarbon emissions mechanisms in Spark Ignition Engines*, SAE Paper 932708, 1993.

30. Zhang, Y. *Possibilities and trends of Ethanol as a transportation fuel*, Proceedings of the 3rd International PSE on EV, Clean Car and Auto Environmental Protection Technology, Beijing, 2001, pp 1- 8.

31. Egnell, R. *Comparison of Heat Release and NO_x Formation in a DI diesel engine running on DME and Diesel fuel*, SAE Paper 2001-01-0651, 2001

32. Chan, S.H., Hoang, D.L. and Zhou, P. L. *Heat Transfer and Chemical Kinetics in the exhaust system of a cold-start engine fitted with a three-way catalytic converter*, Proceedings of the Institute of Mechanical Engineers, Volume 214, Part D, 2000, pp 765 – 777.

33. Iwamoto, M. and Mizuno, N. *NO_x emission control in oxygen-rich exhaust through selective catalytic reduction by hydrocarbons*, Proceedings of the Institute of Mechanical Engineers, Volume 207, Part D, 1993, pp 23 – 33.

34. Bürgler, L., Herzog, P. L. and Zelenka, P. *Strategies to meet US 1994/95 diesel engine Federal Legislation for HSDI diesel engine powered vehicles*, Proceedings of the Institute of Mechanical Engineers, Volume 206, Part D, 1992, pp 47 – 54.

35. McCabe, R. W., King, E. T., Watkins, W. T. and Gandhi, H. S. *Laboratory and vehicle studies of Aldehyde emissions from Alcohol fuels*, SAE Paper 900708, 1990.

36. Wang, H. W., Huang, Z. H., Zhou, L. B., Jiang, D. M. and Yang, Z. L. *Investigation on emission characteristics of a compression ignition engine with oxygenated fuels and exhaust gas recirculation,* Proceedings of the Institute of Mechanical Engineers, Volume 214, Part D, 2000, pp 503 – 508.

37. Hardenburg, H. O. and Schaefer, J. A. *The Use of Ethanol as a fuel for Compression Ignition Engines*, Daimler – Benz AG, Stuttgart, Germany, 1981.

38. Horrocks, R. W. *Light – Duty Diesels – an update on the emissions challenge*, Proceedings of the Institute of Mechanical Engineers, Volume 208, Part D, 1994, pp 289 – 298.

39. Sparis, P. D., Botsaris, P., Karkanis, A. and Tatsios, J. *Three-way catalyst assessment via outlet-inlet temperature measurements: Driving tests*, Proceedings of the Institute of Mechanical Engineers, Volume 211, Part D, 1997, pp 445 – 454.

11 BIBLIOGRAPHY

NASA Facts, National Aeronautics and Space Administration Publication. *Global Warming*, The Earth Science Enterprise Series, NF - 222, December 1999.

Nord, K., Haupt, D., Ahlvik, P. and Egebäck, K. *Particulate Emissions from an Ethanol fuelled Heavy duty Diesel engine equipped with EGR, Catalyst and DPF*, SAE Paper 2004 – 01 – 1987, 2004.

Ahmed, I. *Oxygenated Diesel: Emissions and Performance Characteristics of Ethanol-Diesel Blends in CI Engines,* SAE Paper 2001 – 01 - 2475, 2001.

Rapone, M., Prati, M. V. and Della Ragione, L. *The Effect of Different Traffic Conditions on Catalyst Performance and Exhaust Emissions of a 2 Litre Gasoline Car,* SAE Paper 982708, 1998.

Wagner, T. and Wyszyński, M. L. *Aldehydes and Ketones in engine exhaust emissions – a review,* Proceedings of the Institute of Mechanical Engineers, Volume 210, Part D, 1996, pp 109 – 122.

Dwyer, F. R. and Tanner, J. F. *Business Marketing – Connecting Strategy, Relationships and Learning, 1st International Edition,* McGraw-Hill, Inc., 1999.

Sato, Y., Noda, A. and Sakamoto, T. *Combustion and NO_x Emission Characteristics in a DI Methanol Engine using supercharging with EGR,* SAE Paper 971647, 1997.

Sorenson, S. C. and Mikkelsen, S. *Performance and Emissions of a 0.273 Litre Direct Injection Diesel engine fuelled with Neat Dimethyl Ether,* SAE Paper 950064, 1995.

Chan, S. H. *Thermodynamics in a Turbocharged Direct Injection Diesel Engine,* Proceedings of the Institute of Mechanical Engineers, Volume 212, Part D, 1998, pp 11 – 24.

12 APPENDICES

- Appendix A: Calibration Procedure
- Appendix B: Signal Gas Analyser Operating Procedure
- Appendix C: Selected Discussion Results
- Appendix D: Complete Performance Results
- Appendix E: Complete Emissions Results
- Appendix F: Catalytic Converter Efficiency Sample Calculation

APPENDIX A: CALIBRATION PROCEDURES

Dynamometer (Load Cell) Calibration

- The dynamometer had a static torque arm, which protrudes 40 centimetres from the shaft of the dynamometer.
- Mass pieces were suspended from the static torque arm at random intervals. Prior to suspending the masses, the mass of each was recorded.
- The input voltage to the data acquisition system from the load cell was measured using a voltmeter. These measurements were also noted.
- The process was done for increasing and decreasing loads on the torque arm, after which the increasing and decreasing voltage readings were averaged.
- The calculated torque[1], was then plotted against the average output voltage of the load cell and a linear curve was then fitted to the plotted data points.
- Refer below for the corresponding data points and calibration curve.

[1] Torque = Weight x Torque Arm Length

Table A1: Dynamometer Calibration Data

Mass (kg)	Weight (N)	Torque (Nm)	Voltage max (V)	Voltage min (V)	Average Voltage (V)
0	0	0	3.422	3.417	3.42
0.62	6.08	2.44	3.423	3.483	3.45
1.62	15.90	6.36	3.702	3.795	3.75
2.62	25.71	10.29	4.029	4.089	4.06
2.89	28.36	11.35	4.125	4.217	4.17
3.89	38.17	15.28	4.442	4.528	4.49
4.89	47.99	19.21	4.761	4.827	4.79
5.16	50.64	20.27	4.854	4.842	4.85
6.16	60.45	24.20	5.172	5.249	5.21
7.16	70.26	28.13	5.489	5.542	5.52
7.43	72.91	29.19	5.576	5.655	5.62
8.43	82.72	33.11	5.889	5.956	5.92
9.43	92.54	37.04	6.205	6.254	6.23
9.71	95.28	38.14	6.289	6.357	6.32
10.7	105.00	42.03	6.599	6.654	6.63
11.7	114.81	45.96	6.914	6.945	6.93
11.85	116.28	46.55	6.968	7.017	6.99
14.12	138.56	55.47	7.669	7.706	7.69

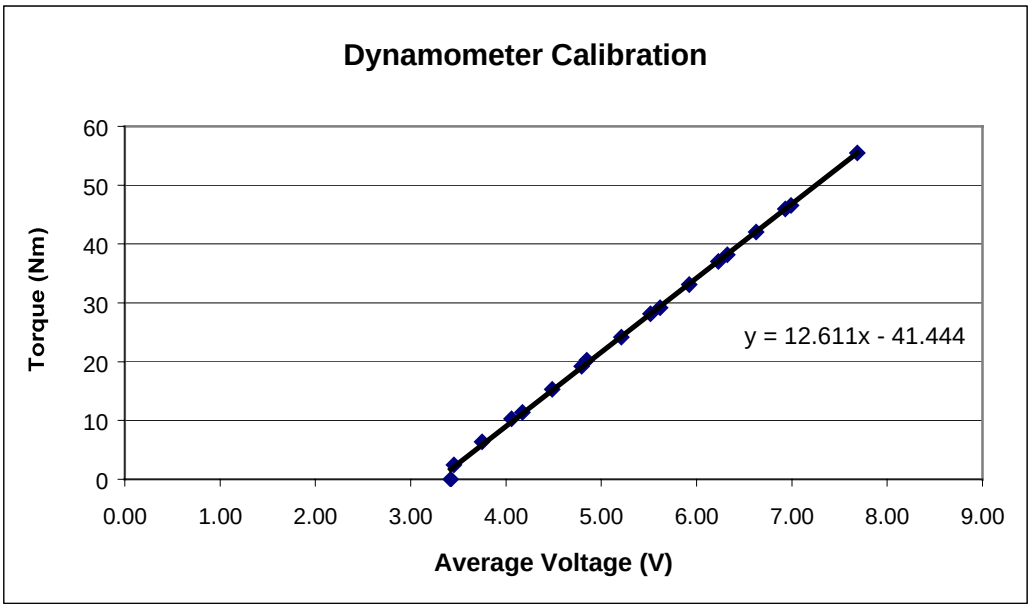

Figure A1: Dynamometer Calibration Curve

Calibration Equation: y = 12.611x – 41.444

Fuel Flow Meter Calibration Procedure

- An electronic scale, a clean beaker and a stopwatch were used for the calibration process.
- Using the scale and the beaker, fuel was bled from the fuel line leaving the flowmeter.
- The stopwatch was used to record the time taken to collect the fuel in the beaker. A two-minute time interval was used.
- For each two-minute time interval, the mass of fuel collected was recorded.
- The input voltage to the data acquisition system was measured by means of a voltmeter and this was also noted.
- The fuel flow was increased for each consecutive reading. The scale of the flowmeter has a maximum value of 12 litres/hour and therefore the readings were randomly selected at approximately 1 litre/hour intervals. These readings were also recorded.
- All the data points i.e. the mass readings and corresponding voltage outputs, were plotted as mass flow rate (g/s) vs. the voltage output of the flowmeter. A linear curve was fitted to the data points from which the calibration equation constants were obtained.
- These results are as shown below:

Table A2: Fuel Flow Calibration Data

Voltage max (V)	Voltage min (V)	Average Voltage (V)	Fuel mass (g)	Time (s)	Massflow (g/s)	Fuel flow (l/h)
0.026	0.03	0.028	0	120	0	0
0.543	0.545	0.544	40.8	120	0.34	1
1.025	1.08	1.052	80.8	120	0.673	2.2
1.541	1.553	1.547	118.4	120	0.986	3.2
1.863	1.883	1.873	144.6	120	1.205	4
2.264	2.272	2.268	177.2	120	1.476	5
2.274	2.754	2.513	213.4	120	1.778	6
3.192	3.209	3.2	248.5	120	2.07	7
3.75	3.761	3.755	292.2	120	2.435	8.4
3.99	3.999	3.994	309.7	120	2.58	9
4.588	4.599	4.593	352.2	120	2.935	10.4
4.91	4.921	4.915	183.7	60	3.061	11.2

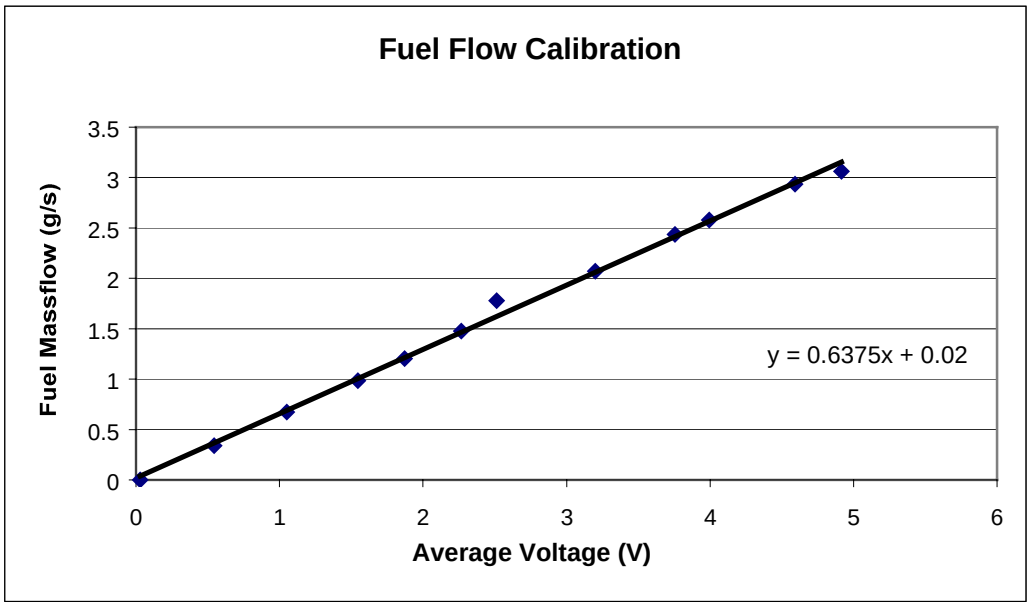

Figure A2: Fuel Flow Calibration Curve

Calibration Equation: $y = 0.6375x + 0.02$

Airflow Calibration

- The airflow was measured using an orifice plate fitted at one end of a large drum above the engine.
- While the engine was idling, pressure readings were taken across the orifice plate.
- These values were displayed in millimetres of water.
- Corresponding voltage readings were taken, as the speed of the engine increased, the airflow intake would increase accordingly and both the airflow and the voltage readings were taken.
- These results were tabulated and plotted.
- A linear curve was fitted through the points to determine the calibration constants to be imported into the computer programme.
- Refer below for the data and the calibration curve.

Table A3: Air Flow Calibration Data

Pressure (mmH_2O)	**Voltage (V)**
25.7	1.349
44.6	2.231
54.9	2.752
58.9	2.99
56.6	3.289
74.7	3.745
78.9	3.963
85.9	4.308

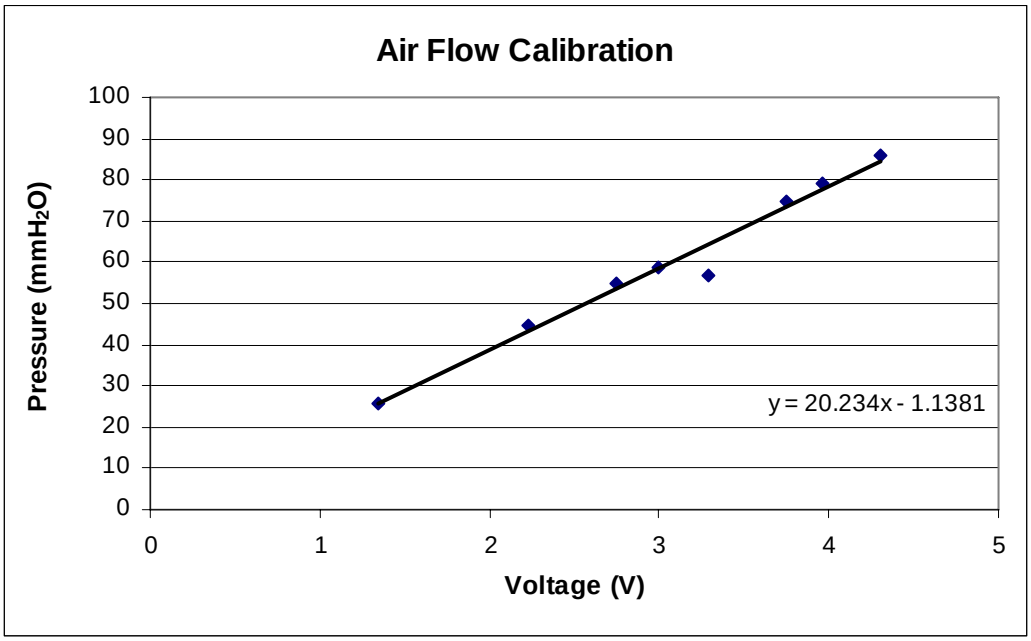

Figure A3: Air Flow Calibration Curve

Calibration Equation: y = 20.234x – 1.1381

Thermocouple Calibration

- The calibration procedure for all the temperature channels were identical.
- As illustration, the ambient air temperature thermocouple calibration will be explained.
- The thermocouple was placed in a beaker of tap water, and left for a while until a constant initial temperature was recorded on the multimeter.
- The corresponding voltage output was read off the data acquisition system.
- The data was tabulated and a corresponding calibration curve was plotted.
- For each temperature channel, a calibration curve was produced, refer below for each temperature channels' tabulated data and calibration curve.

i. Ambient Air Temperature
ii. Inlet Catalytic Converter Temperature
iii. Outlet Catalytic Converter Temperature
iv. Air Intake Temperature
v. Inlet Water Temperature
vi. Outlet Water Temperature.

i. Ambient Air Temperature Calibration

Table A4: Ambient Air Temperature Calibration Data

Temp (deg. C)	Voltage (V)
13.2	0.152
15.7	0.173
27.1	0.338
35.1	0.355
46	0.47
55	0.583
66.2	0.684
75.3	0.773
85	0.879
93.1	0.962
94.6	0.971
94.7	0.972

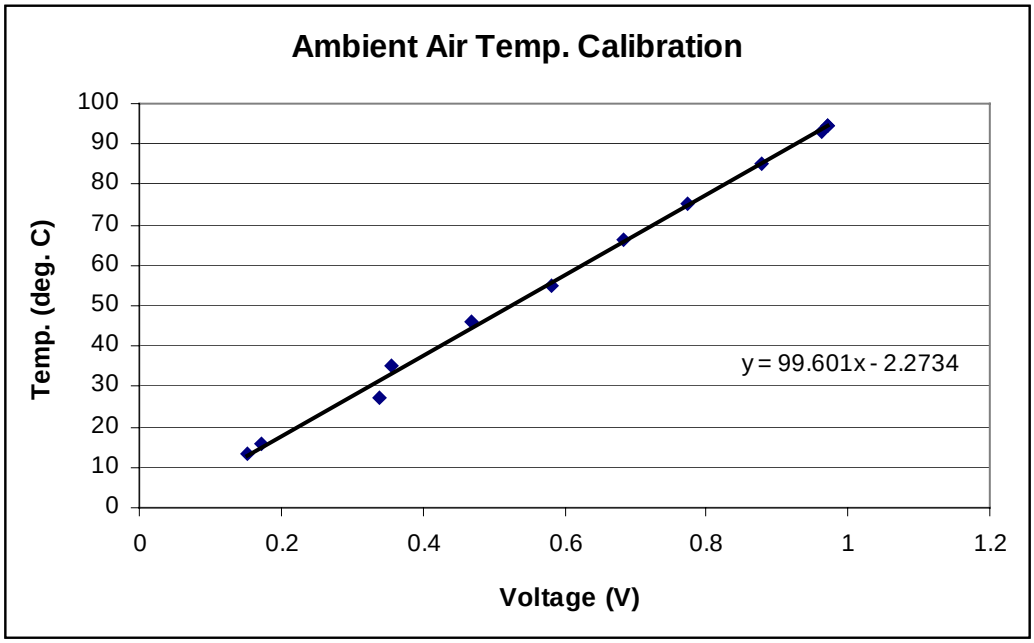

Figure A4: Ambient Air Temperature Calibration Curve

Calibration Equation: $y = 99.601x - 2.2734$

ii. Inlet Catalytic Converter Temperature Calibration

Table A5: Inlet Catalytic Converter Temperature Calibration Data

Temp (deg. C)	Voltage (V)
14	0.16
16	0.156
26.2	0.248
35	0.355
45	0.46
56.1	0.551
65	0.658
75	0.758
85.9	0.864
94.5	0.949

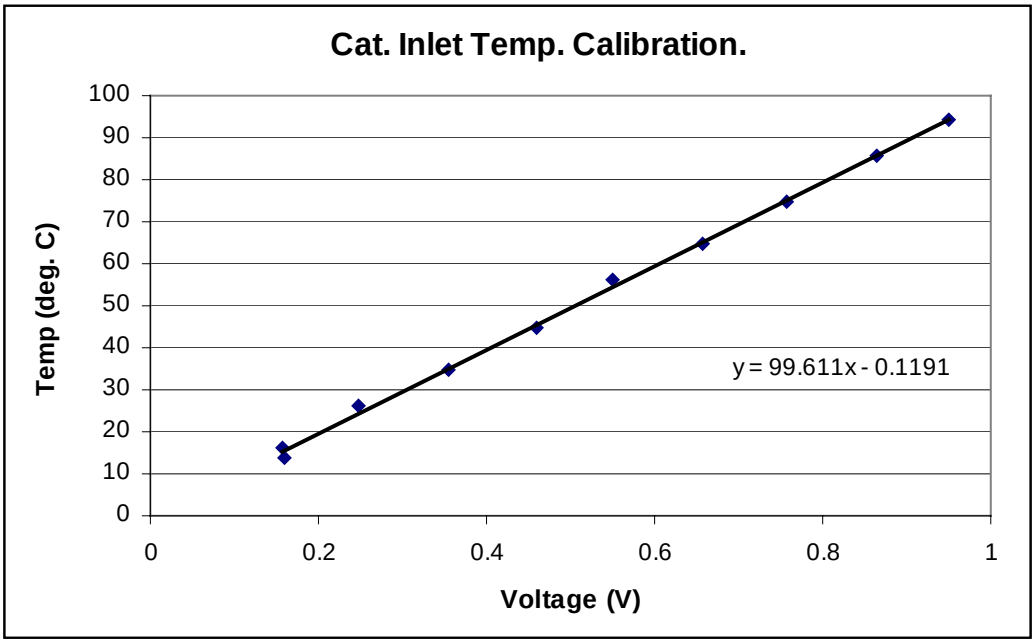

Figure A5: Inlet Catalytic Converter Temperature Calibration Curve

Calibration Equation: y = 99.611x – 0.1191

iii. Outlet Catalytic Converter Temperature Calibration

Table A6: Outlet Catalytic Converter Temperature Calibration Data

Temp (deg. C)	Voltage (V)
18.5	0.219
25	0.292
35	0.39
45	0.49
55	0.595
65	0.697
75	0.8
85	0.9
93	0.967

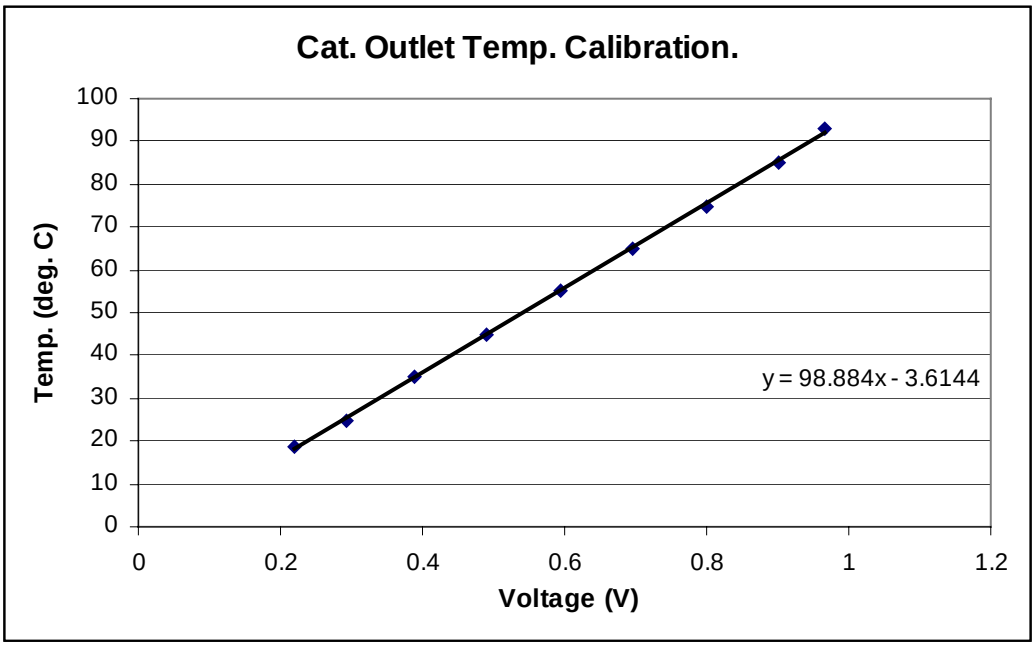

Figure A6: Outlet Catalytic Converter Temperature Calibration Curve

Calibration Equation: $y = 98.884x - 3.6144$

iv. Air Intake Temperature Calibration

Table A7: Air Intake Temperature Calibration Data

Temp (deg. C)	Voltage (V)
13.1	0.172
16.2	0.195
25	0.294
35	0.393
45	0.499
55	0.596
65	0.698
67.8	0.722
69.2	0.737
75	0.804
85	0.898
94.5	0.986

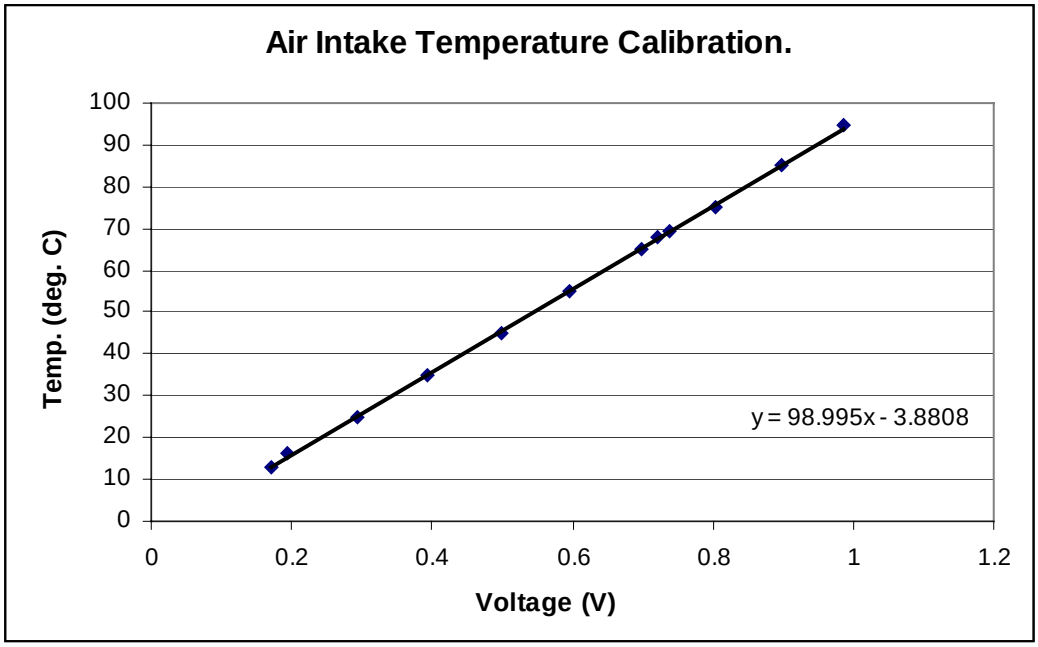

Figure A7: Air Intake Temperature Calibration Curve

Calibration Equation: y = 98.995x – 3.8808

v. Inlet Water Temperature Calibration

Table A8: Inlet Water Temperature Calibration Data

Temp (deg. C)	Voltage (V)
13.8	0.215
23.5	0.278
30	0.346
40	0.439
50	0.541
60	0.639
70	0.74
80	0.85
90	0.938
94.7	0.984

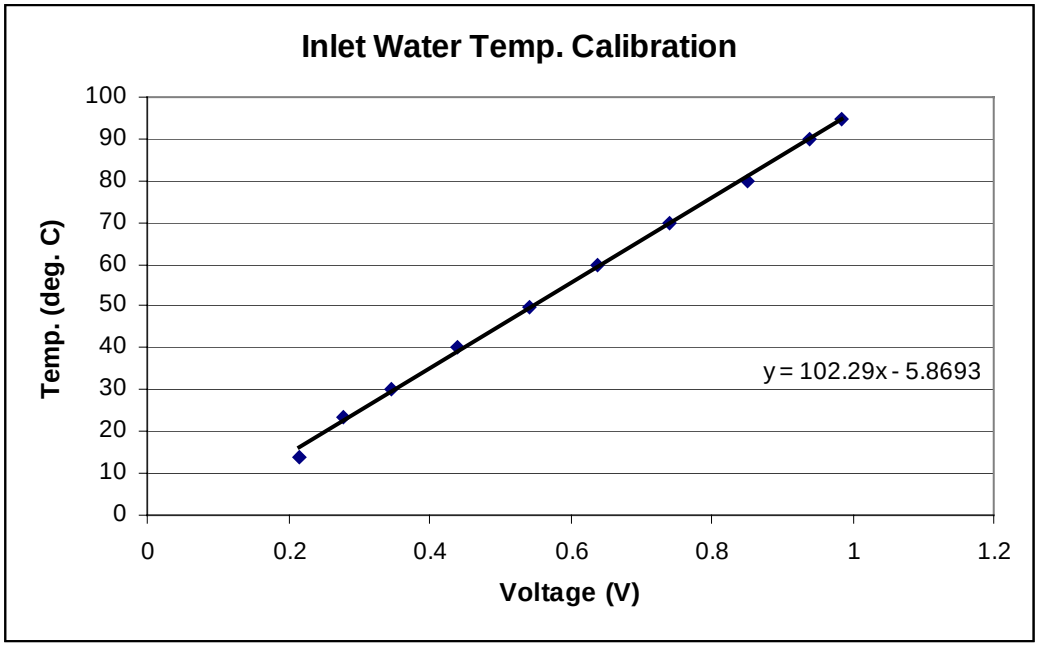

Figure A8: Inlet Water Temperature Calibration Curve

Calibration Equation: $y = 102.29x - 5.8693$

vi. Outlet Water Temperature Calibration

Table A9: Outlet Water Temperature Calibration Data

Temp (deg. C)	Voltage (V)
14.3	0.197
21.6	0.218
30	0.309
40	0.407
50	0.517
60	0.618
70	0.718
80	0.806
90	0.907
94.7	0.948

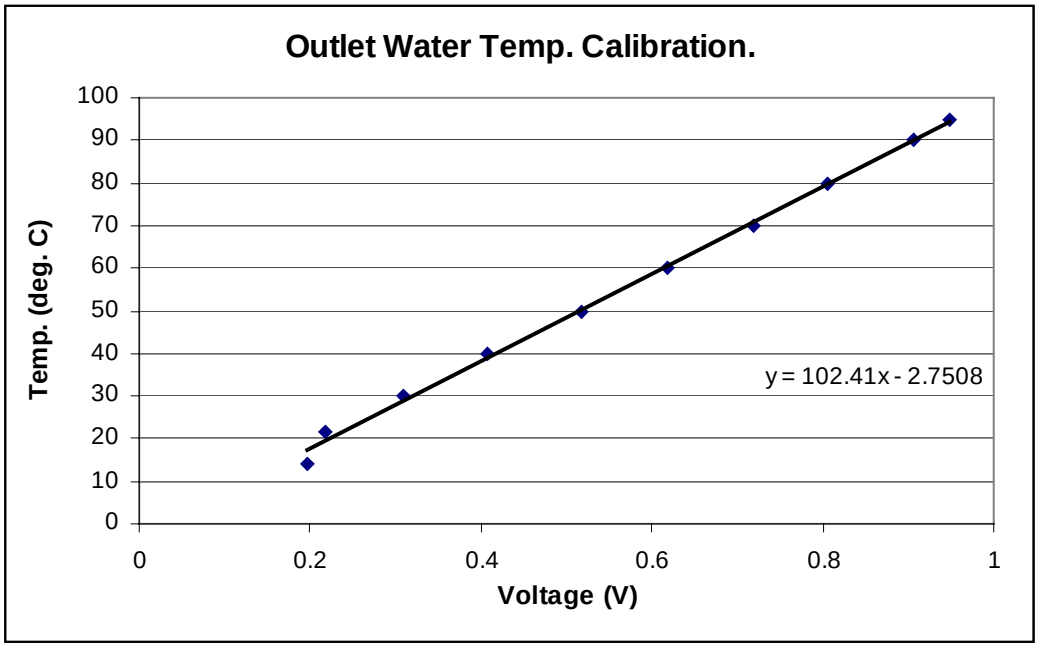

Figure A9: Outlet Water Temperature Calibration Curve

Calibration Equation: $y = 102.41x - 2.7508$

APPENDIX B:

SIGNAL GAS ANALYSER

Start up, Calibration and Shut down Procedure.

The procedures to follow are for the Signal Series III Analyser. For each test session these procedures have to be performed to ensure good working order of the analyser.

Start up Procedure

- Switch on the main power supply at the wall plug. The pumps will start up, the oven lights will burn and the fans will turn on.
- Switch on all the analyser units (CO, CO_2, NO_X, THC and cooler/dryer). The switches to these analysers are on the front panels of each unit.
- All the display panels (LCD – liquid crystal display) will now be lit up and go into standby mode.
- A Status warning will show on all the LCD displays of the analyser units.
- Press the STATUS button and then press the PAGE DOWN button twice. Do this for all the analyser units.
- The window being displayed at this stage on all of the units should be the "Health Check Window", which gives the operating status of the analyser by listing all the operating variables. A flashing variable alerts the user to a problem with that specific variable. Upon start up, usually temperature, pressure and/or the flow variables will be flashing.
- Allow all units to stand as they are for approximately 45 minutes to an hour, to allow the temperatures in the NO_X (oven and converter temperatures) and THC (oven and catalyst temperatures) analysers to reach the operating temperatures required. The variables will stop flashing to indicate operating status.
- The THC unit has a FLAME temperature that will only increase to operating temperature when the gas bottle for "fuel" is opened, and the THC analyser is in SAMPLE mode (procedure to follow).

Analyser Requirements and Gas Cylinder Contents

Table B1: Gas Cylinder Contents

Analyser	Cylinder Label	Cylinder Contents	Description
THC	Span	1000 ppm Methane (CH_4) in air	Large
	Fuel	40 % H_2 in Helium	Large
NO_X	Zero	99.9 % Nitrogen	Large
	Span	3030 ppm NO in N_2	Large
CO	Zero	99.9 % Nitrogen	Large
	Span	8.0 % in air	Large
CO_2	Zero	99.9 % Nitrogen	Large
	Span	15.6 % CO_2 in air	Large

Calibration Procedure

- Once the NO_X oven and converter and, the THC oven and catalyst temperatures have reached their required levels (i.e. no longer flashing), then the next phase can begin. Double check to see that the above four temperature variables on the LCD panels are not flashing.
- Now press the ESC (Escape) button on all the analysers.
- Press the SAMPLE button (the button below the LCD screen aligned with SAMP on the screen) on all the analysers. This will change the analyser units from STANDBY to SAMPLE mode.
- Open the THC FUEL and the NO_X ZERO (nitrogen) gas cylinders. **NB:** Do not turn the regulator dials. All the cylinders have regulators that have been pre-set because of the flows and pressures required by each analyser. The main cylinder valve is the one to be opened.
- Open the laboratory's supply of compressed air by turning the valve halfway (45^o). The compressed air passes through three filters, namely particulate, water and oil.
- The CO_2, CO and NO_X analysers should now be in SAMPLE mode without any STATUS warnings.
- The THC analyser will give a STATUS FAIL warning, as the analyser prepares for SAMPLE and FLAME temperature are still low. Press the STATUS button and then press the PAGE DOWN button twice to go back to the "Health Check" window. During this phase the Gas Flow is zero. Once the analyser is in SAMPLE mode, the Gas flow value will no longer be zero and the FLAME temperature would have been reached.
- The THC analyser will try (5) times to light the flame and increase the FLAME temperature. If all five tries have elapsed and the flame temperature is still flashing, then press ESC to exit the "Health Check" window. Now press STOP and then SAMPLE to commence the igniting process again.
- Once the FLAME temperature on the THC analyser is ready, calibration may begin.
- When calibrating, each analyser is calibrated separately, first starting with the THC, then the NO_X, followed by the CO_2 and lastly the CO analyser.

- Open a single SPAN cylinder corresponding to the analyser to be calibrated. Now press CAL button (the button below the LCD screen aligned with CAL on the screen) on the analyser, and wait for the calibration process to complete before moving onto the next analyser.
- The calibration process on the analyser first processes ZERO calibration and then proceeds to SPAN calibration, the analyser then reads "SAMPLE SETTLING" on the LCD screen, afterwhich it returns to SAMPLE mode. Once this is done, the calibration process is complete.
- If a STATUS WARNING is given after the calibration, one or some of the variables have not reached their required status for calibration. Proceed to the "Health Check" window and establish which variables are flashing.
- Most often it is the gas flow variables which flash, this means insufficient gas is flowing through the analyser. **NB:** Adjust the regulator knobs on the gas cylinder until the variable stops flashing, do this carefully. ***This is the only time that the regulator knobs should be turned otherwise not.*** Re-calibrate the analyser.
- After calibration of a single analyser the SPAN cylinder should be shut. Only the ZERO and FUEL cylinders remain open throughout until testing is complete.
- Now check to see that the calibration process was successful. This is done by pressing the STATUS button, and then the PAGE DOWN button once. The effective range should lie between –100% and +100% for all the analyser units.
- *Note: If the STATUS WARNING appears, due to N_2 gas flow on the CO or CO_2 while calibration occurs is because N_2 is used by both CO and CO_2 at the same time, it should stop flashing after calibration, if it does not the regulator knob should be adjusted until it stops on both analysers.*
- If all the preceding procedures have been completed, refer to the table which follows, to check that all the variables on the 'Health Check Window' are within the given ranges.

VARIABLE RANGES FOR EACH ANALYSER

If the variable values are not within range, the analyser will give a "STATUS WARNING" error message.

Table B2: Variable Ranges for CO_2 Analyser

Analyser	Variable	Normal Reading	Upper Limit Reading	Lower Limit Reading
CO_2 7000 M	Amb. Temp.	40 °C	58 °C	0 °C
	Sens Status	GOOD	FAULT [1]	
	P. S. Status	GOOD	FAULT [1]	
	N_2 Pressure	0.14 bar	Check footnote # **2**	
	Gas Flow	0.7 l/m	1.0 l/m	0 l/m
	Measure 147.9 POH			

Table B3: Variable Ranges for CO Analyser

Analyser	Variable	Normal Reading	Upper Limit Reading	Lower Limit Reading
CO 7000 M	Amb. Temp.	38 °C	58 °C	0 °C
	Sens Status	GOOD	FAULT [1]	
	P. S. Status	GOOD	FAULT [1]	
	N_2 Pressure	0.18 bar	Check footnote # **2**	
	Gas Flow	0.5 l/m	1.0 l/m	0 l/m

Table B4: Variable Ranges for THC Analyser

Analyser	Variable	Normal Reading	Upper Limit Reading	Lower Limit Reading
THC 3000 HM	Oven Temp.	191 °C	281 °C	201 °C
	Cat. Temp.	636 °C	680 °C	584 °C
	Amb. Temp.	40 °C	58 °C	0 °C
	Flame Temp.	380 °C	Check footnote # **3**	
	P. S. Status	GOOD	FAULT [1]	
	H. T. Status	GOOD	FAULT [1]	
	Fuel Pressure	0.88 bar	1.1 bar	0.6 bar
	Ref. Pressure	0.45 bar	0.5 bar	0.4 bar
	Gas Flow	1.7 l/m	3.0 l/m	0.5 l/m

Table B5: Variable Ranges for NO_x Analyser

Analyser	Variable	Normal Reading	Upper Limit Reading	Lower Limit Reading
NO_X 4000 VM	Oven Temp.	195 °C	205 °C	175 °C
	Conv. Temp.	383 °C	400 °C [4]	360 °C
	Amb. Temp.	39 °C	60 °C	0 °C
	P. S. Status	GOOD	FAULT [1]	
	H. T. Status	GOOD	FAULT [1]	
	O_3 Pressure	1.06 bar	1.25 bar [5]	0.75 bar
	Vac. Pressure	0.05 bar	0.05 bar	0.01 bar
	Gas Flow	1.9 l/m	5.0 l/m	0.5 l/m

NOTE: IN THE ABOVE TABLES, THE NORMAL READING FOR ALL THE VARIABLES INDICATES PERFECT OPERATION, HOWEVER SHOULD THE VARIABLE READINGS VARY SLIGHTLY BUT ARE STILL WITHIN RANGE; NO ERROR MESSAGES WILL APPEAR.

Notes

1 The analyser will give a "FAULT" warning, refer to the manual for help or contact a Signal technician.

2 The user manuals do not specify upper and lower limits. If a STATUS WARNING appears on the LCD display then it means insufficient N_2 is flowing through the analyser. Page down to the "Health Check Window" and the N_2 gas flow variable will be flashing. Adjust the regulator on the N_2 gas cylinder until the N_2 variable on the "Health Check Window" stops flashing; check that this is achieved on both the CO and CO_2 analyser.

3 Flashing of this variable will commence, which indicates the temperature is too high or too low.

4 This value is not given in the operating manuals.

5 The optimum O_3 pressure is 1.00 bar and 1.25 bar is an estimated Upper limit.

PROBLEMS WITH CALIBRATION

Should any problems occur with the calibration of the analyser, a CAL WARNING error message will appear on the LCD screen. The following procedure should be used.

- On the front panel of the analyser, with the error message, press the STATUS button and then the PAGE DOWN button once. The effective range for any of the variables should lie between –100% and +100%. The error message will have appeared if one or more of the variables lie outside the –100% and +100% limits. Find out which of these is causing the fault.
- Check to see that the SPAN/ZERO main valve on the cylinder for that variable is open. If it is closed, open it and recalibrate.
- If the valve is open already, then place the analyser in the correct mode to read the value of the cylinder contents. To get the analyser in this mode, while in the sample mode screen, press the PAGE DOWN button once and notice on the LCD screen, the SPAN and ZERO will appear on the bottom of the screen.
- If the SPAN is giving the problem, press SPAN and wait for the analyser to read the contents of the SPAN of the cylinder. Once this value has been established, adjust the potentiometer on the inside of the analyser front panel (see user manual for location).
- By adjusting the potentiometer, the value being read by the analyser can be adjusted to read the value of the cylinder.
- Once this has been done, close the front panel, place the analyser back into SAMPLE mode and recalibrate.

Shut Down

- Once testing is complete for the day, remove the heated line from the exhaust connection.
- Allow the analyser to “air out” for 30 minutes to an hour.
- Press STOP on the front panels of all the analyser units. The analyser units will now begin to purge themselves (120 seconds).
- When this is complete, the units will all be in STANDBY mode.
- Now close all the cylinder main valves and do not tamper with the regulator valves.
- Turn off the air supply.
- Switch off the front panel power switches.
- Switch off the wall plug power.

Signal Gas Analyser Software: Saving Data

Once the PC has been started up in Windows and the Analyser unit has been calibrated, then the software programme may be set up to capture the test data. The programme used is Signal and this measures the emissions of the engine in a graphical mode by calculating the average concentration of the emissions and plotting the results as the test is being performed. If anything were to go wrong during a test then one would immediately notice this by the change in pattern of the graphs being plotted. The emissions measured are the Nitrogen Oxide (NO_X), Total unburned hydrocarbons (THC), Carbon monoxide (CO) and Carbon dioxide (CO_2). The following procedure briefly outlines the steps involved whilst performing tests with emissions.

- Double click on the **Signal** icon (torch flame picture) to open the **Signal Exhaust Analyser Software** operating window.
- Click the **Test** button on the menu bar, scroll to click the **Start New Test** button. This will then open the **Start New Test Information Window.**
- Here the test name is entered as before, and the operator can select the number of data points. For this research project either 50 or 100 points were chosen depending on the time available for each test.
- The **Reading Interval** was set at every **1 second.**
- Clicking the icon with the little folders on it can choose the directory, to which the data is to be saved. Here select the path C:\SignalProgam\Signaldata\{Folder}. The entry {Folder} is to be replaced with whichever folder the data is to be stored in. This is usually the date of the test and the test number, eg. 2702, which reads 27th of November and test number two for that day.
- The operating window shows all concentrations graphically and numerically as the test is being run.
- For each new test, the new test name needs to be re-entered in order to be stored for further use.
- The captured data may be viewed in Notepad at a later stage for further use.

APPENDIX C: SELECTED DISCUSSION RESULTS

Performance Results for Diesel fuel:

Table C1: Performance Data for Diesel Fuel

1550 rpm Diesel (Standard 3 hole injectors)										
Test Number	**Speed (rpm)**	**Load (Nm)**	**Brake Power (kW)**	**Equivalence Ratio**	**Mechanical Efficiency (%)**	**Fuel Conversion Efficiency (%)**	**Before Cat. Temp. (^{o}C)**	**Max Cylinder Pressure (kPa)**	**Ignition Delay (degrees)**	**bsfc (kg/J)**
di050202	1550.9	7.2	1.17	0.278	30.0	9.1	178.8	4793.7	11.8	2.50E-07
di050207	1550.9	14.3	2.33	0.310	47.3	16.5	210.2	5061.2	11.2	1.40E-07
di050208	1553.7	22.5	3.66	0.367	59.2	22.0	246.9	5505.5	10.4	1.00E-07
di050215	1553.7	28.0	4.56	0.413	66.7	24.5	285.2	5680.7	10.6	9.40E-08
di050218	1553.7	37.0	6.02	0.498	74.6	27.0	338.1	6017.5	9.8	8.50E-08
di050220	1556.5	43.0	7.01	0.546	76.6	29.0	385.6	6262.3	9.0	7.90E-08
di050224	1556.5	50.4	8.22	0.610	83.8	30.4	431.2	6466.7	9.8	7.50E-08
di050230	1556.5	55.0	8.96	0.720	84.1	29.7	492.4	6672.1	9.8	8.10E-08
di050233	1559.3	58.4	9.53	0.813	85.3	28.6	532.6	6765.4	9.4	8.50E-08
di050239	1559.3	61.6	10.05	0.974	88.6	24.3	560.1	6855.5	9.0	9.70E-08
di050240	1559.3	62.1	10.15	1.190	91.7	19.7	548.3	6767.7	8.8	1.20E-07

Performance Results for Ethanol/DME fuel:

Table C2: Performance Data for Ethanol/DME Fuel 3 Hole

1550 rpm Ethanol/DME (Standard 3 hole injectors)										
Test Number	**Speed (rpm)**	**Load (Nm)**	**Brake Power (kW)**	**Equivalence Ratio**	**Mechanical Efficiency (%)**	**Fuel Conversion Efficiency (%)**	**Before Cat. Temp. (^{o}C)**	**Max Cylinder Pressure (kPa)**	**Ignition Delay (degrees)**	**bsfc (kg/J)**
ed180200	1550.9	8.7	1.42	0.375	33.5	12.4	184.2	5341.1	9.8	2.80E-07
ed180207	1553.7	13.7	2.22	0.427	46.0	15.9	179.8	5404.8	7.6	2.10E-07
ed180209	1550.9	22.7	3.69	0.472	58.6	23.3	225.4	5878.7	5.2	1.50E-07
ed180215	1553.7	29.1	4.73	0.527	64.2	25.5	255.8	6195.4	6.4	1.30E-07
ed180217	1550.9	37.8	6.14	0.622	72.9	26.7	309.7	6395.7	4.2	1.30E-07
ed180220	1553.7	44.0	7.16	0.687	77.3	27.5	350.3	6522.5	3.4	1.20E-07
ed180225	1548.1	50.3	8.16	0.781	82.5	26.5	392.5	7147.7	0.0	1.30E-07
ed180235	1553.7	58.6	9.53	0.871	89.8	27.0	476.3	6684.4	0.0	1.30E-07
ed180239	1553.7	61.3	9.97	0.941	95.0	25.7	520.4	6201.0	2.0	1.30E-07

Performance Results for Ethanol/DME fuel continued:

Table C3: Performance Data for Ethanol/DME Fuel 4 Hole

1550 rpm Ethanol/DME (Standard 4 hole injectors)										
Test Number	**Speed (rpm)**	**Load (Nm)**	**Brake Power (kW)**	**Equivalence Ratio**	**Mechanical Efficiency (%)**	**Fuel Conversion Efficiency (%)**	**Before Cat. Temp. (^{o}C)**	**Max Cylinder Pressure (kPa)**	**Ignition Delay (degrees)**	**bsfc (kg/J)**
ed130302	1548.1	8.1	1.31	0.404	30.9	10.0	182.7	5080.2	7.6	3.40E-07
ed130306	1548.1	14.3	2.33	0.420	56.9	17.4	213.6	5146.2	8.4	2.00E-07
ed130311	1550.9	22.1	3.59	0.504	69.2	20.7	243.0	5302.5	4.8	1.60E-07
ed130314	1550.9	28.7	4.66	0.555	73.1	23.6	268.5	5712.4	5.6	1.40E-07
ed130318	1550.9	35.6	5.78	0.642	79.6	24.1	308.7	6078.5	4.6	1.40E-07
ed140304	1553.7	44.1	7.17	0.712	83.0	26.2	351.3	6606.4	6.0	1.30E-07
ed140306	1553.7	51.6	8.40	0.800	91.7	26.3	403.7	6616.3	5.6	1.30E-07
ed140311	1556.5	54.6	8.89	0.832	93.1	26.5	438.5	6684.6	5.6	1.30E-07
ed140315	1556.5	58.5	9.53	0.883	96.7	26.4	472.8	6816.8	4.6	1.30E-07

Emissions Results for Diesel fuel:

Table C4: Emissions Data for Diesel Fuel Before Catalytic Converter

1550 rpm Diesel (Standard 3 hole injectors): Before Catalytic Converter								
Test Number	**Brake Power (kW)**	**Equivalence Ratio**	***s*NO_x (ppm/kW)**	***s*THC (ppm/kW)**	***s*CO (ppm/kW)**	***s*CO_2 (ppm/kW)**	***s*O_2 (%/kW)**	***s*(Smoke) (HSU/kW)**
di050202	1.17	0.278	385.3	1192.5	473.5	28563.8	15.0	0.9
di050207	2.33	0.297	315.5	632.9	297.6	18354.4	7.0	0.9
di050208	3.66	0.367	294.1	499.2	183.7	14163.7	4.1	0.5
di050215	4.56	0.413	318.5	418.9	99.7	13575.1	3.0	0.7
di050218	6.02	0.498	309.0	371.6	78.5	12312.1	1.9	1.3
di050220	6.99	0.568	304.0	262.3	93.4	12374.9	1.4	1.6
di050224	8.31	0.697	274.1	278.7	144.9	12027.0	0.9	2.8
di050230	8.96	0.720	243.8	270.1	292.8	12114.8	0.7	3.7
di050233	9.53	0.813	206.9	294.0	678.9	12175.1	0.4	5.5
di050239	10.05	0.974	125.0	330.7	2079.1	12223.0	0.0	7.9

Emissions Results for Diesel fuel continued:

Table C5: Emissions Data for Diesel Fuel After Catalytic Converter

1550 rpm Diesel (Standard 3 hole injectors): After Catalytic Converter								
Test Number	**Brake Power (kW)**	**Equivalence Ratio**	***s*NO_x (ppm/kW)**	***s*THC (ppm/kW)**	***s*CO (ppm/kW)**	***s*CO_2 (ppm/kW)**	***s*O_2 (%/kW)**	***s*(Smoke) (HSU/kW)**
di050202	1.17	0.278	441.3	1120.1	530.8	28176.6	14.8	0.9
di050207	2.33	0.297	268.6	240.9	17.0	19171.6	6.9	0.9
di050208	3.66	0.367	249.4	177.2	14.9	15061.8	3.7	0.5
di050215	4.56	0.413	244.6	115.3	12.3	14178.7	2.9	0.7
di050218	6.02	0.498	241.5	132.6	11.6	12865.5	1.8	1.3
di050220	6.99	0.568	262.9	91.7	11.2	12640.0	1.4	1.6
di050224	8.31	0.697	270.0	95.4	16.6	12634.2	0.8	2.8
di050233	9.53	0.813	207.4	42.3	14.7	12992.6	0.4	5.5
di050239	10.05	0.974	99.5	36.1	20.1	15110.5	0.0	7.9

Emissions Results for Ethanol/DME fuel:

Table C6: Emissions Data for Ethanol/DME Fuel 3 Hole Before Catalytic Converter

1550 rpm Ethanol/DME (Standard 3 hole injectors): Before Catalytic Converter							
Test Number	**Brake Power (kW)**	**Equivalence Ratio**	**sNO_x (ppm/kW)**	**sTHC (ppm/kW)**	**sCO (ppm/kW)**	**sCO_2 (ppm/kW)**	**sO_2 (%/kW)**
ed180200	1.42	0.375	84.3	2354.1	3908.1	21705.8	12.6
ed180207	2.22	0.427	69.6	1535.9	2209.3	18041.6	8.1
ed180209	3.69	0.472	142.1	712.4	1062.0	11906.4	4.3
ed180215	4.73	0.527	143.2	418.8	596.3	11621.3	3.0
ed180217	6.14	0.622	196.1	293.7	396.5	11409.7	2.0
ed180220	7.16	0.687	205.6	184.7	325.0	11168.5	1.5
ed180225	8.16	0.781	204.6	192.2	279.6	11614.1	1.0
ed180235	9.53	0.871	165.1	117.6	444.3	11484.2	0.7
ed180239	9.97	0.941	133.1	71.2	857.7	11719.5	0.4

Emissions Results for Ethanol/DME fuel continued:

Table C7: Emissions Data for Ethanol/DME Fuel 3 Hole After Catalytic Converter

1550 rpm Ethanol/DME (Standard 3 hole injectors): After Catalytic Converter							
Test Number	**Brake Power (kW)**	**Equivalence Ratio**	**sNO_x (ppm/kW)**	**sTHC (ppm/kW)**	**sCO (ppm/kW)**	**sCO_2 (ppm/kW)**	**sO_2 (%/kW)**
ed180200	1.42	0.375	115.2	559.0	101.9	28062.0	12.68
ed180207	2.22	0.427	128.9	225.8	103.5	21127.8	7.87
ed180209	3.69	0.472	111.2	52.8	20.3	13981.4	4.07
ed180215	4.73	0.527	122.8	28.4	14.8	12723.8	2.96
ed180217	6.14	0.622	149.6	26.8	11.0	12289.7	1.94
ed180220	7.16	0.687	189.3	14.7	11.7	11948.4	1.41
ed180225	8.16	0.781	193.7	14.4	12.7	12093.1	0.93
ed180235	9.53	0.871	164.3	24.9	7.4	11948.3	0.63
ed180239	9.97	0.941	145.8	13.1	14.8	12597.9	0.40

Emissions Results for Ethanol/DME fuel continued:

Table C8: Emissions Data for Ethanol/DME Fuel 4 Hole Before Catalytic Converter

1550 rpm Ethanol/DME (Standard 4 hole injectors): Before Catalytic Converter							
Test Number	**Brake Power (kW)**	**Equivalence Ratio**	**sNO_x (ppm/kW)**	**sTHC (ppm/kW)**	**sCO (ppm/kW)**	**sCO_2 (ppm/kW)**	**sO_2 (%/kW)**
ed130302	1.31	0.404	201.3	1411.9	3682.3	27451.7	16.1
ed130306	2.33	0.420	176.9	662.9	1647.8	18282.2	9.0
ed130311	3.59	0.504	170.1	364.5	834.3	14876.2	4.1
ed130314	4.66	0.555	146.2	249.1	600.6	12374.7	3.0
ed130318	5.78	0.642	200.4	176.5	409.2	12865.4	2.0
ed140304	7.17	0.712	215.4	136.8	326.5	12485.7	1.3
ed140306	8.40	0.800	172.5	106.5	311.1	12448.9	0.8
ed140311	9.05	0.832	145.4	62.2	486.8	12773.4	0.6
ed140315	9.53	0.883	119.0	68.3	891.5	12869.1	0.4

Emissions Results for Ethanol/DME fuel continued:

Table C9: Emissions Data for Ethanol/DME Fuel 4 Hole After Catalytic Converter

1550 rpm Ethanol/DME (Standard 4 hole injectors): After Catalytic Converter							
Test Number	**Brake Power (kW)**	**Equivalence Ratio**	**sNO_x (ppm/kW)**	**sTHC (ppm/kW)**	**sCO (ppm/kW)**	**sCO_2 (ppm/kW)**	**sO_2 (%/kW)**
ed130302	1.31	0.404	178.2	149.5	893.2	26019.9	16.1
ed130306	2.33	0.420	169.2	131.2	53.0	18374.9	9.0
ed130311	3.59	0.504	137.3	81.0	48.4	14780.4	5.6
ed130314	4.66	0.555	151.9	34.6	18.5	12863.1	4.6
ed130318	5.78	0.642	192.8	7.6	32.8	13326.4	3.6
ed140304	7.17	0.712	191.7	28.9	13.3	12232.0	2.9
ed140306	8.40	0.800	167.4	11.6	33.0	12053.9	2.3
ed140311	9.05	0.832	148.9	19.9	10.7	12187.5	2.3
ed140315	9.53	0.883	117.9	18.2	16.0	12365.6	2.2

Catalytic Converter Efficiency:

Table C10: Diesel Fuel Efficiency

1550 rpm Diesel (Standard 3 hole injectors)						
Test Number	**Equivalence Ratio**	**Before Cat Temp (^oC)**	**After Cat Temp (^oC)**	**NO_x (%)**	**THC (%)**	**CO (%)**
di050202	0.278	178.8	214	(-14.52)[1]	6.07	(-12.10)
di050207	0.297	208.7	267	14.87	61.94	94.30
di050208	0.367	246.9	311	15.20	64.50	91.91
di050215	0.413	285.2	337	23.20	72.48	87.66
di050218	0.498	338.1	381	21.86	64.31	85.28
di050220	0.568	387.1	426	13.52	65.04	88.04
di050224	0.697	437.6	478	1.48	65.76	88.57
di050230	0.720	492.4	540	(-2.20)	75.97	96.44
di050233	0.813	532.6	605	(-0.25)	85.62	97.83
di050239	0.974	560.1	785	20.40	89.10	99.04

[1] Efficiency percentages resulting in negative values were omitted for the purposes of trend analysis in the discussion. The reason being, these values could be due to experimental fluctuations in the emissions data captured.

Catalytic Converter Efficiency continued:

Table C11: Ethanol/DME 3 Hole Fuel Efficiency

1550 rpm Ethanol/DME (Standard 3 hole injectors)						
Test Number	**Equivalence Ratio**	**Before Cat Temp (^{o}C)**	**After Cat Temp (^{o}C)**	**NO_x (%)**	**THC (%)**	**CO (%)**
ed180200	0.375	184.2	194.5	(-36.66)	76.25	97.39
ed180207	0.427	184.7	239.7	(-85.10)	85.30	95.32
ed180209	0.472	225.4	311.7	21.69	92.59	98.09
ed180215	0.527	255.8	323.4	14.21	93.23	97.52
ed180217	0.622	309.7	359.7	23.71	90.88	97.24
ed180220	0.687	350.3	393.0	7.93	92.06	96.40
ed180225	0.781	392.5	439.0	5.34	92.49	95.47
ed180235	0.871	476.3	512.0	0.45	78.84	98.33
ed180239	0.941	520.4	575.0	(-9.51)	81.57	98.28

Catalytic Converter Efficiency continued:

Table C12: Ethanol/DME 4 Hole Fuel Efficiency

1550 rpm Ethanol/DME (Standard 4 hole injectors)						
Test Number	**Equivalence Ratio**	**Before Cat Temp (^{o}C)**	**After Cat Temp (^{o}C)**	**NO_x (%)**	**THC (%)**	**CO (%)**
ed130302	0.404	182.7	217.3	11.50	89.41	75.74
ed130306	0.420	213.6	250.6	4.37	80.20	96.79
ed130311	0.504	243	293.1	19.30	77.78	94.20
ed130314	0.555	268.5	307.6	(-3.88)	86.10	96.92
ed130318	0.642	308.7	346.4	3.78	95.69	91.98
ed140304	0.712	351.3	368.3	11.00	78.89	95.93
ed140306	0.800	403.7	415.9	2.99	89.09	89.93
ed140311	0.832	444.9	457.6	(-2.38)	68.08	97.79
ed140315	0.883	472.8	508.5	0.97	73.27	98.21

APPENDIX D: COMPLETE PERFORMANCE RESULTS

Performance Results for Diesel fuel:

Table D1: Performance Data for Diesel Fuel

1550 rpm Diesel (Standard 3 hole injectors) 210 bar										
Test Number	**Fuel Type**	**Speed (rpm)**	**Load (Nm)**	**Brake Power (W)**	**Indicated Work/Cycle (J)**	**Indicated Power (W)**	**imep (kPa)**	**bmep (kPa)**	**bmep (MPa)**	**Equivalence Ratio**
di050202	Diesel	1550.9	7.2	1169.7	150.8	3898.0	226.8	68.0	0.068	0.278
di050207	Diesel	1550.9	14.3	2330.0	190.7	4929.0	286.8	135.5	0.136	0.31
di050208	Diesel	1553.7	22.5	3656.0	238.7	6180.0	358.9	212.3	0.212	0.367
di050215	Diesel	1553.7	28.0	4558.0	264.1	6838.0	397.1	264.7	0.265	0.413
di050218	Diesel	1553.7	37.0	6021.0	311.7	8073.0	468.8	349.6	0.350	0.498
di050220	Diesel	1556.5	43.0	7005.0	352.5	9144.0	530.0	406.1	0.406	0.546
di050224	Diesel	1556.5	50.4	8220.0	378.2	9811.0	568.7	476.5	0.477	0.61
di050230	Diesel	1556.5	55.0	8962.0	410.8	10656.0	617.7	519.5	0.520	0.72
di050233	Diesel	1559.3	58.4	9531.0	430.2	11179.0	646.9	551.5	0.552	0.813
di050239	Diesel	1559.3	61.6	10054.0	436.5	11343.0	656.3	581.8	0.582	0.974
di050240	Diesel	1559.3	62.1	10154.0	425.7	11062.0	640.1	587.0	0.587	1.19

Performance Results for Diesel fuel continued:

Table D2: Performance Data for Diesel Fuel

1550 rpm Diesel (Standard 3 hole injectors) 210 bar								
Test Number	**Air/Fuel Ratio**	**Air Flow (g/s)**	**Fuel Flow (g/s)**	**Fuel Conversion Efficiency (%)**	**isfc (kg/J)**	**bsfcf (kg/J)**	**Injection Point (degrees)**	**Ignition Point (degrees)**
di050202	53.88	15.824	0.29	9.1	7.50E-08	2.50E-07	345.9	357.7
di050207	48.32	15.693	0.32	16.5	6.60E-08	1.40E-07	346.3	357.5
di050208	40.90	15.575	0.38	22.0	6.20E-08	1.00E-07	346.7	357.1
di050215	36.22	15.482	0.43	24.5	6.30E-08	9.40E-08	346.7	357.3
di050218	30.01	15.348	0.51	27.0	6.30E-08	8.50E-08	346.70	356.60
di050220	27.41	15.213	0.55	29.0	6.10E-08	7.90E-08	346.70	355.70
di050224	24.48	15.186	0.62	30.4	6.30E-08	7.50E-08	345.90	355.70
di050230	20.73	15.118	0.73	29.7	6.80E-08	8.10E-08	346.10	355.90
di050233	18.41	14.968	0.81	28.6	7.30E-08	8.50E-08	345.90	355.30
di050239	15.34	14.955	0.97	24.3	8.60E-08	9.70E-08	345.70	354.70
di050240	12.56	14.858	1.18	19.7	1.10E-07	1.20E-07	346.10	354.90

Performance Results for Diesel fuel continued:

Table D3: Performance Data for Diesel Fuel

1550 rpm Diesel (Standard 3 hole injectors) 210 bar							
Test Number	**Ignition Delay (degrees)**	**Pressure Atm. (kPa)**	**Before Cat. Temp. (°C)**	**After Cat. Temp. (°C)**	**Max Pressure (kPa)**	**Position of Pmax (degrees)**	**Max Temp. (K)**
di050202	11.8	82.0	178.8	214.0	4793.7	363.4	1257.4
di050207	11.2	81.5	210.2	264.0	5061.2	364.0	1336.4
di050208	10.4	81.1	246.9	311.0	5505.5	363.8	1494.2
di050215	10.6	81.1	285.2	337.0	5680.7	363.0	1536.5
di050218	9.8	81.1	338.1	381.0	6017.5	366.8	1663.5
di050220	9.0	81.1	385.6	424.0	6262.3	366.4	1769.9
di050224	9.8	81.1	431.2	472.0	6466.7	365.8	1791.9
di050230	9.8	81.1	492.4	540.0	6672.1	366.4	1879.6
di050233	9.4	81.1	532.6	605.0	6765.4	365.8	1931.1
di050239	9.0	83.5	560.1	785.0	6855.5	365.8	1961.6
di050240	8.8	83.5	548.3	779.0	6767.7	366.0	1966.5

Performance Results for Diesel fuel continued:

Table D4: Performance Data for Diesel Fuel

1550 rpm Diesel (Standard 3 hole injectors) 210 bar					
Test Number	**Position of Tmax (degrees)**	**Mechanical Efficiency (%)**	**ITE (%)**	**BTE (%)**	**Volumetric Efficiency (%)**
di050202	370.2	30.0	30.4	9.1	92.1
di050207	370.2	47.3	34.8	16.5	91.3
di050208	369.4	59.2	37.2	22.0	90.4
di050215	368.8	66.7	36.7	24.5	89.9
di050218	368.0	74.6	36.2	27.0	89.1
di050220	369.2	76.6	37.8	29.0	88.2
di050224	368.6	83.8	36.3	30.4	88
di050230	370.8	84.1	33.5	28.2	87.6
di050233	370.2	85.3	31.5	26.9	86.6
di050239	371.8	88.6	26.7	23.7	86.5
di050240	370.4	91.7	21.4	19.7	86

Performance Results for Ethanol/DME fuel:

Table D5: Performance Data for Ethanol/DME Fuel

1550 rpm Ethanol/DME (Standard 3 hole injectors) 210 bar								
Test Number	**Fuel Type**	**Speed (rpm)**	**Load (Nm)**	**Brake Power (W)**	**Indicated Work/Cycle (J)**	**Indicated Power (W)**	**imep (kPa)**	**bmep (kPa)**
ed180200	Ethanol/DME	1550.9	8.7	1419.7	163.9	4237.0	246.5	82.6
ed180207	Ethanol/DME	1553.7	13.7	2223.7	186.7	4836.0	280.8	129.1
ed180209	Ethanol/DME	1550.9	22.7	3689.9	243.8	6302.0	366.6	214.7
ed180215	Ethanol/DME	1553.7	29.1	4728.4	284.3	7361.0	427.4	274.6
ed180217	Ethanol/DME	1550.9	37.8	6140.1	325.6	8418.0	489.7	357.2
ed180220	Ethanol/DME	1553.7	44.0	7163.0	357.8	9266.0	538.1	416.0
ed180225	Ethanol/DME	1548.1	50.3	8156.0	383.1	9885.0	576.1	475.3
ed180235	Ethanol/DME	1553.7	58.6	9527.0	409.9	10613.0	616.3	553.3
ed180239	Ethanol/DME	1553.7	61.3	9968.0	408.8	10492.0	609.4	578.9

Performance Results for Ethanol/DME fuel continued:

Table D6: Performance Data for Ethanol/DME Fuel

1550 rpm Ethanol/DME (Standard 3 hole injectors) 210 bar							
Test Number	**Equivalence Ratio**	**Ethanol Equi. Ratio**	**DME Equi. Ratio**	**Air/Fuel Ratio**	**Air Flow (g/s)**	**Fuel Flow (g/s)**	**Additive Flow (g/s)**
ed180200	0.375	0.084	0.108	39.96	15.615	0.20	0.19
ed180207	0.427	0.12	0.108	32.73	15.641	0.29	0.19
ed180209	0.472	0.148	0.109	28.49	15.389	0.35	0.19
ed180215	0.527	0.186	0.109	24.39	15.376	0.44	0.19
ed180217	0.622	0.252	0.111	19.45	15.16	0.59	0.19
ed180220	0.687	0.298	0.112	17.09	15.078	0.69	0.19
ed180225	0.781	0.369	0.112	14.35	14.982	0.86	0.19
ed180235	0.871	0.438	0.113	12.44	14.886	1.01	0.19
ed180239	0.941	0.49	0.114	11.29	14.802	1.12	0.19

Performance Results for Ethanol/DME fuel continued:

Table D7: Performance Data for Ethanol/DME Fuel

1550 rpm Ethanol/DME (Standard 3 hole injectors) 210 bar						
Test Number	**% of Additive (DME)**	**Fuel Conversion Efficiency (%)**	**isfc (kg/J)**	**bsfcf (kg/J)**	**ITE (%)**	**BTE (%)**
ed180200	47.9	12.4	9.20E-08	2.80E-07	36.5	12.4
ed180207	39.20	15.9	9.90E-08	2.10E-07	34.1	15.9
ed180209	34.70	23.3	8.60E-08	1.50E-07	39.3	23.3
ed180215	29.70	25.5	8.60E-08	1.30E-07	39.3	25.6
ed180217	24.00	26.7	9.30E-08	1.30E-07	36.4	26.8
ed180220	21.20	27.5	9.50E-08	1.20E-07	35.4	27.6
ed180225	17.90	26.5	1.10E-07	1.30E-07	31.9	26.5
ed180235	15.70	27.0	1.10E-07	1.30E-07	29.9	27.0
ed180239	14.30	25.7	1.30E-07	1.30E-07	26.1	25.0

Performance Results for Ethanol/DME fuel continued:

Table D8: Performance Data for Ethanol/DME Fuel

1550 rpm Ethanol/DME (Standard 3 hole injectors) 210 bar							
Test Number	**Injection Point (degrees)**	**Ignition Point (degrees)**	**Ignition Delay (degrees)**	**DME Ignition (press)**	**DME Ignition 1st (begin)**	**DME Ignition 1st (end)**	**1st Stage DME (degrees)**
ed180200	346.9	356.7	9.8	334.2	331.4	332.4	1.0
ed180207	348.1	355.7	7.6	332.8	333.0	333.0	0.0
ed180209	348.1	353.3	5.2	333.0	331.8	332.8	1.0
ed180215	347.9	354.3	6.4	334.0	330.8	333.2	2.4
ed180217	348.1	352.3	4.2	330.0	329.0	330.4	1.4
ed180220	348.30	351.70	3.4	330.6	328.8	329.0	0.2
ed180225	353.30	353.30	0.0	330.4	330.6	331.4	0.8
ed180235	353.30	353.30	0.0	330.2	327.6	330.0	2.4
ed180239	353.10	355.10	2.0	334.0	326.2	326.2	0.0

Performance Results for Ethanol/DME fuel continued:

Table D9: Performance Data for Ethanol/DME Fuel

1550 rpm Ethanol/DME (Standard 3 hole injectors) 210 bar							
Test Number	**Energy at DME 1st (J)**	**1st Stage Energy (J)**	**DME Ignition 2nd (begin)**	**DME Ignition 2nd (end)**	**2nd Stage DME (degrees)**	**Energy at DME 2nd (J)**	**2nd Stage Energy (J)**
ed180200	-22.0	3.7	348.8	353.4	4.6	-1.9	37.5
ed180207	4.6	0.0	352.8	353.8	1.0	65.1	12.7
ed180209	-29.5	7.2	345.2	350.4	5.2	-3.4	77.4
ed180215	-10.3	14.8	353.2	359.6	6.4	114.8	220.5
ed180217	-10.1	11.1	351.4	364.8	13.4	118.7	350.9
ed180220	3.7	4.0	351.2	364.0	12.8	140.0	351.3
ed180225	-8.8	7.7	350.4	355.4	5.0	112.3	189.6
ed180235	-47.8	13.0	349.4	370.6	21.2	95.9	544.9
ed180239	-30.5	0.0	349.8	366.0	16.2	105.4	426.2

Performance Results for Ethanol/DME fuel continued:

Table D10: Performance Data for Ethanol/DME Fuel

1550 rpm Ethanol/DME (Standard 3 hole injectors) 210 bar						
Test Number	**Pressure Atm. (kPa)**	**Before Cat. Temp. (°C)**	**Max Pressure (kPa)**	**Position of Pmax (degrees)**	**Max Temp. (K)**	**Position of Tmax (degrees)**
ed180200	83.0	184.2	5341.1	362.0	1336.9	367.6
ed180207	83.0	179.8	5404.8	362.0	1372.0	367.8
ed180209	83.0	225.4	5878.7	364.4	1527.7	368.6
ed180215	82.5	255.8	6195.4	363.8	1614.2	368.0
ed180217	82.5	309.7	6395.7	364.2	1707.3	368.8
ed180220	82.5	350.3	6522.5	365.0	1752.8	371.0
ed180225	82.5	392.5	7147.7	364.8	1902.0	369.4
ed180235	83.0	476.3	6684.4	365.2	1843.9	371.6
ed180239	83.0	520.4	6201.0	364.6	1707.7	372.2

Performance Results for Ethanol/DME fuel continued:

Table D11: Performance Data for Ethanol/DME Fuel

1550 rpm Ethanol/DME (Standard 3 hole injectors) 210 bar				
Test Number	**Mechanical Efficiency (%)**	**Volumetric Efficiency (%)**	**Ambient Temp. (°C)**	**After Cat. Temp. (°C)**
ed180200	33.5	90.8	25.6	194.5
ed180207	46.0	90.8	26.1	239.7
ed180209	58.6	89.5	26.5	305.7
ed180215	64.2	89.3	26.7	323.4
ed180217	72.9	88.2	26.9	359.7
ed180220	77.3	87.6	27.1	393.0
ed180225	82.5	87.3	27.4	439.0
ed180235	89.8	86.4	28.5	512.0
ed180239	95.0	85.9	32.7	575.0

Performance Results for Ethanol/DME fuel continued:

Table D12: Performance Data for Ethanol/DME Fuel (4 Hole injector)

1550 rpm Ethanol/DME (Standard 4 hole injectors) 210 bar								
Test Number	**Fuel Type**	**Speed (rpm)**	**Load (Nm)**	**Brake Power (W)**	**Indicated Work/Cycle (J)**	**Indicated Power (W)**	**imep (kPa)**	**bmep (kPa)**
ed130302	Ethanol/DME	1548.1	8.1	1307.3	164.1	4234.2	246.8	76.2
ed130306	Ethanol/DME	1548.1	14.3	2325.6	158.3	4085.2	238.1	135.5
ed130311	Ethanol/DME	1550.9	22.1	3589.9	200.7	5187.1	301.8	208.8
ed130314	Ethanol/DME	1550.9	28.7	4660.0	246.6	6374.0	370.8	271.1
ed130318	Ethanol/DME	1550.9	35.6	5780.1	280.9	7260.1	422.4	336.3
ed140304	Ethanol/DME	1553.7	44.1	7173.0	333.9	8646.5	502.1	416.5
ed140306	Ethanol/DME	1553.7	51.6	8395.3	353.4	9152.5	531.5	487.5
ed140311	Ethanol/DME	1556.5	54.6	8892.1	368.0	9546.6	553.4	515.5
ed140315	Ethanol/DME	1556.5	58.5	9534.5	379.9	9855.7	571.3	552.7

Performance Results for Ethanol/DME fuel continued:

Table D13: Performance Data for Ethanol/DME Fuel (4 Hole injector)

1550 rpm Ethanol/DME (Standard 4 hole injectors) 210 bar							
Test Number	**Equivalence Ratio**	**Ethanol Equi. Ratio**	**DME Equi. Ratio**	**Air/Fuel Ratio**	**Air Flow (g/s)**	**Fuel Flow (g/s)**	**Additive Flow (g/s)**
ed130302	0.404	0.106	0.107	35.36	15.800	0.26	0.19
ed130306	0.420	0.113	0.110	33.65	15.349	0.27	0.19
ed130311	0.504	0.170	0.110	25.98	15.322	0.40	0.19
ed130314	0.555	0.205	0.111	22.70	15.228	0.48	0.19
ed130318	0.642	0.267	0.111	18.61	15.146	0.63	0.19
ed140304	0.712	0.318	0.112	16.23	15.078	0.74	0.19
ed140306	0.800	0.384	0.112	13.89	15.024	0.89	0.19
ed140311	0.832	0.408	0.112	13.21	14.982	0.95	0.19
ed140315	0.883	0.447	0.112	12.23	14.969	1.04	0.19

Performance Results for Ethanol/DME fuel continued:

Table D14: Performance Data for Ethanol/DME Fuel (4 Hole injector)

1550 rpm Ethanol/DME (Standard 4 hole injectors) 210 bar						
Test Number	**% of Additive (DME)**	**Fuel Conversion Efficiency (%)**	**isfc (kg/J)**	**bsfcf (kg/J)**	**ITE (%)**	**BTE (%)**
ed130302	41.90	10.0	1.10E-07	3.40E-07	31.9	9.9
ed130306	41.10	17.4	1.10E-07	2.00E-07	30.2	17.2
ed130311	31.80	20.7	1.10E-07	1.60E-07	29.6	20.8
ed130314	27.90	23.6	1.10E-07	1.40E-07	32.0	23.6
ed130318	23.00	24.1	1.10E-07	1.40E-07	30.0	23.9
ed140304	20.20	26.2	1.10E-07	1.30E-07	31.3	26.2
ed140306	17.30	26.3	1.20E-07	1.30E-07	28.5	26.3
ed140311	16.50	26.5	1.20E-07	1.30E-07	28.3	26.6
ed140315	15.30	26.4	1.20E-07	1.30E-07	27.1	26.4

Performance Results for Ethanol/DME fuel continued:

Table D15: Performance Data for Ethanol/DME Fuel (4 Hole injector)

1550 rpm Ethanol/DME (Standard 4 hole injectors) 210 bar							
Test Number	**Injection Point (degrees)**	**Ignition Point (degrees)**	**Ignition Delay (degrees)**	**DME Ignition (press)**	**DME Ignition 1st (begin)**	**DME Ignition 1st (end)**	**1st Stage DME (degrees)**
ed130302	346.7	354.3	7.6	334.2	325.2	325.2	0.0
ed130306	347.5	355.9	8.4	333.4	331.8	333.8	2.0
ed130311	347.7	352.5	4.8	331.8	332.0	332.0	0.0
ed130314	347.9	353.5	5.6	331.2	330.8	333.2	2.4
ed130318	348.3	352.9	4.6	331.0	330.2	330.2	0.0
ed140304	346.7	352.7	6.0	330.2	330.0	330.6	0.6
ed140306	346.5	352.1	5.6	331.2	331.4	331.4	0.0
ed140311	346.3	351.9	5.6	330.0	330.2	330.2	0.0
ed140315	346.7	351.3	4.6	329.6	329.4	330.0	0.6

Performance Results for Ethanol/DME fuel continued:

Table D16: Performance Data for Ethanol/DME Fuel (4 Hole injector)

1550 rpm Ethanol/DME (Standard 4 hole injectors) 210 bar							
Test Number	**Energy at DME 1st (J)**	**1st Stage Energy (J)**	**DME Ignition 2nd (begin)**	**DME Ignition 2nd (end)**	**2nd Stage DME (degrees)**	**Energy at DME 2nd (J)**	**2nd Stage Energy (J)**
ed130302	2.8	0.0	347.0	353.4	5.4	20.3	51.4
ed130306	-16.4	13.3	348.2	353.4	5.2	2.0	47.3
ed130311	-46.1	0.0	346.2	352.0	5.8	-33.8	57.3
ed130314	-20.8	15.0	351.4	358.0	6.6	62.0	221.3
ed130318	-1.3	0.0	344.6	363.8	19.2	13.2	449.8
ed140304	2.1	4.9	351.8	358.4	6.6	115.4	290.9
ed140306	58.6	0.0	351.2	364.6	13.4	148.8	464.0
ed140311	-47.6	0.0	351.8	362.6	10.8	68.5	406.6
ed140315	-14.2	4.8	350.4	367.2	16.8	111.6	497.6

Performance Results for Ethanol/DME fuel continued:

Table D17: Performance Data for Ethanol/DME Fuel (4 Hole injector)

1550 rpm Ethanol/DME (Standard 4 hole injectors) 210 bar						
Test Number	**Pressure Atm. (kPa)**	**Before Cat. Temp. (°C)**	**Max Pressure (kPa)**	**Position of Pmax (degrees)**	**Max Temp. (K)**	**Position of Tmax (degrees)**
ed130302	85.4	182.7	5080.2	359.6	1261.6	365.0
ed130306	83.0	213.6	5146.2	360.4	1302.2	366.0
ed130311	83.0	243.0	5302.5	363.8	1368.8	364.8
ed130314	83.0	268.5	5712.4	363.4	1482.5	367.8
ed130318	83.0	308.7	6078.5	363.8	1593.2	368.2
ed140304	83.5	351.3	6606.4	363.2	1743.8	367.8
ed140306	83.5	403.7	6616.3	364.4	1762.6	369.0
ed140311	83.5	438.5	6684.6	365.4	1789.4	368.6
ed140315	83.5	472.8	6816.8	365.2	1838.0	371.6

Performance Results for Ethanol/DME fuel continued:

Table D18: Performance Data for Ethanol/DME Fuel (4 Hole injector)

1550 rpm Ethanol/DME (Standard 4 hole injectors) 210 bar				
Test Number	**Mechanical Efficiency (%)**	**Volumetric Efficiency (%)**	**Ambient Temp. (°C)**	**After Cat. Temp. (°C)**
ed130302	30.9	92.1	26.2	217.34
ed130306	56.9	89.5	27.0	250.56
ed130311	69.2	89.1	27.6	293.10
ed130314	73.1	88.6	27.9	307.64
ed130318	79.6	88.1	28.4	346.37
ed140304	83.0	87.6	26.8	368.31
ed140306	91.7	87.2	27.5	415.90
ed140311	93.1	86.8	28.4	457.61
ed140315	96.7	86.8	28.9	508.54

APPENDIX E: COMPLETE EMISSIONS RESULTS

Diesel fuel Emissions before and after catalytic converter:

Table E1: Emissions Data for Diesel Fuel – Before Catalytic Converter

1550 rpm Diesel (Standard 3 hole injectors) 210 bar: Before Catalytic Converter											
Load (Nm)	**Equivalence Ratio**	**Bef. Cat Temp (^oC)**	**After Cat Temp (^oC)**	**Brake Power (W)**	**Air-fuel Ratio**	**NO_x (ppm)**	**THC (ppm)**	**CO (ppm)**	**CO_2 (ppm)**	**O_2 (%)**	**Smoke (HSU)**
7.164	0.278	178.8	214	1169.7	53.88	450.7	1394.84	553.88	33411.06	17.5	1
14.241	0.297	208.7	267	2329.8	50.24	735.16	1474.5	693.38	42762	16.2	2
22.481	0.367	246.9	311	3656.4	40.9	1075.32	1825.44	671.56	51788	14.9	2
27.978	0.413	285.2	337	4558.1	36.22	1451.9	1909.43	454.267	61876.67	13.6	3
36.894	0.498	338.1	381	6020.9	30.01	1860.66	2237.22	472.46	74130	11.3	8
42.887	0.568	387.1	426	6985.1	26.28	2123.66	1832.22	652.14	86440	9.5	11.5
51.028	0.697	437.6	478	8310	21.47	2277.36	2316.28	1203.86	99944	7.2	23.5
54.942	0.72	492.4	540	8962.4	20.73	2184.72	2420.8	2624.08	108578	6	33
58.332	0.813	532.6	605	9531.4	18.41	1971.68	2801.76	6470.7	116046	4.2	52
61.555	0.974	560.1	785	10054.3	15.34	1257.08	3325.3	20904.2	122894	0	79.5

Diesel fuel Emissions before and after catalytic converter continued:

Table E2: Emissions Data for Diesel Fuel – After Catalytic Converter

1550 rpm Diesel (Standard 3 hole injectors) 210 bar: After Catalytic Converter											
Load (Nm)	**Equivalence Ratio**	**Bef. Cat Temp (^oC)**	**After Cat Temp (^oC)**	**Brake Power (W)**	**Air-fuel Ratio**	**NO_x (ppm)**	**THC (ppm)**	**CO (ppm)**	**CO_2 (ppm)**	**O_2 (%)**	**Smoke (HSU)**
7.164	0.278	178.8	214	1169.7	53.88	218.1	1310.16	620.92	32958.2	17.3	1
14.241	0.297	208.7	267	2329.8	50.24	625.86	561.16	39.54	44666	16.0	2
22.481	0.367	246.9	311	3656.4	40.9	911.82	648.08	54.32	55072	13.4	2
27.978	0.413	285.2	337	4558.1	36.22	1115.12	525.46	56.04	64628	13.2	3
36.894	0.498	338.1	381	6020.9	30.01	1453.98	798.52	69.56	77462	11.0	8
42.887	0.568	387.1	426	6985.1	26.28	1836.58	640.56	77.98	88292	9.5	11.5
51.028	0.697	437.6	478	8310	21.47	2243.6	793.1	137.58	104990	7.0	23.5
54.942	0.72	492.4	540	8962.4	20.73	2232.86	581.62	93.38	115766	-	33
58.332	0.813	532.6	605	9531.4	18.41	1976.64	402.84	140.36	123838	3.4	52
61.555	0.974	560.1	785	10054.3	15.34	1000.62	362.54	201.68	151926	0	79.5

Ethanol/DME fuel Emissions before and after catalytic converter:

Table E3: Emissions Data for Ethanol/DME Fuel – Before Catalytic Converter

1550 rpm Ethanol/DME (Standard 3 hole injectors) 210 bar: Before Catalytic Converter										
Load (Nm)	**Equivalence Ratio**	**Bef. Cat Temp (^oC)**	**After Cat Temp (^oC)**	**Brake Power (W)**	**Air-fuel Ratio**	**NO_x (ppm)**	**THC (ppm)**	**CO (ppm)**	**CO_2 (ppm)**	**O_2 (%)**
8.39	0.375	184.2	194.5	1419.7	37.03	119.63	3342.1	5548.33	30815.67	17.9
13.43	0.427	184.7	239.7	2223.7	29.65	154.854	3415.38	4912.72	40119.2	18
22.609	0.472	225.4	311.7	3689.9	24.54	524.167	2628.53	3918.54	43933.33	16
28.899	0.527	255.8	323.4	4728.4	21.43	677.1	1980.367	2819.77	54950	14.2
37.76	0.622	309.7	359.7	6140.1	17.49	1204.08	1803.533	2434.7	70056.67	12.1
43.7	0.687	350.3	393	7163	15.56	1472.9	1322.9	2328.067	79999.96	10.5
50.604	0.781	392.5	439	8155.6	13.25	1668.9	1567.9	2280.6	94720	8.5
58.505	0.871	476.3	512	9527.4	11.6	1572.71	1120.5	4233.25	109414.3	6.2
61.366	0.941	520.4	575	9968.3	10.58	1327.07	709.6	8549.7	116823.33	4.2

Ethanol/DME fuel Emissions before and after catalytic converter continued:

Table E4: Emissions Data for Ethanol/DME Fuel – After Catalytic Converter

1550 rpm Ethanol/DME (Standard 3 hole injectors) 210 bar: After Catalytic Converter										
Load (Nm)	**Equivalence Ratio**	**Bef. Cat Temp (^oC)**	**After Cat Temp (^oC)**	**Brake Power (W)**	**Air-fuel Ratio**	**NO_x (ppm)**	**THC (ppm)**	**CO (ppm)**	**CO_2 (ppm)**	**O_2 (%)**
8.39	0.375	184.2	194.5	1419.7	37.03	163.49	793.68	144.6	39839.6	18
13.43	0.427	184.7	239.7	2223.7	29.65	286.64	502.212	230.15	46981.82	17.5
22.609	0.472	225.4	311.7	3689.9	24.54	410.48	194.667	74.933	51590	15
28.899	0.527	255.8	323.4	4728.4	21.43	580.867	134.167	69.833	60163.33	14
37.76	0.622	309.7	359.7	6140.1	17.49	918.567	164.533	67.267	75460	11.9
43.997	0.687	350.3	393	7163	15.56	1356.03	105.06	83.9	85586.67	10.1
50.604	0.781	392.5	439	8155.6	13.25	1579.86	117.76	103.233	98626.67	7.6
58.505	0.871	476.3	512	9527.4	11.6	1565.57	237.13	70.7	113836.67	6
61.366	0.941	520.4	575	9968.3	10.58	1453.26	130.8	147.267	125580	4

Ethanol/DME fuel Emissions before and after catalytic converter continued:

Table E5: Emissions Data for Ethanol/DME Fuel (4 Hole injector) – Before Catalytic Converter

1550 rpm Ethanol/DME (Standard 4 hole injectors) 210 bar: Before Catalytic Converter										
Load (Nm)	**Equivalence Ratio**	**Bef. Cat Temp (^oC)**	**After Cat Temp (^oC)**	**Brake Power (W)**	**Air-fuel Ratio**	**NO_x (ppm)**	**THC (ppm)**	**CO (ppm)**	**CO_2 (ppm)**	**O_2 (%)**
8.1	0.404	182.7	217.34	1307.3	29.6	263.16	1845.82	4813.9	35887.67	21
14.3	0.42	213.6	250.56	2325.6	28.26	411.4	1541.74	3832.1	42517	21
22.1	0.504	243	293.1	3589.9	22.64	610.66	1308.64	2994.92	53404	14.6
28.7	0.555	268.5	307.64	4660	20.1	681.52	1161	2798.68	57666	13.8
35.6	0.642	308.7	346.37	5780.1	16.18	1158.267	1020.1	2365.27	74363.33	11.3
44.1	0.712	351.3	368.31	7173.8	14.84	1545.53	981.167	2342.4	89570	9
51.6	0.8	403.7	415.9	8395.3	12.86	1448.54	894	2611.96	104512	7
55.5	0.832	444.9	457.61	9052.7	12.24	1316.54	563.38	4407.12	115634	5
58.5	0.883	472.8	508.54	9534.5	11.41	1135	651	8500	122700	4

Ethanol/DME fuel Emissions before and after catalytic converter continued:

Table E6: Emissions Data for Ethanol/DME Fuel (4 Hole injector) – After Catalytic Converter

1550 rpm Ethanol/DME (Standard 4 hole injectors) 210 bar: After Catalytic Converter										
Load (Nm)	**Equivalence Ratio**	**Bef. Cat Temp (^oC)**	**After Cat Temp (^oC)**	**Brake Power (W)**	**Air-fuel Ratio**	**NO_x (ppm)**	**THC (ppm)**	**CO (ppm)**	**CO_2 (ppm)**	**O_2 (%)**
8.1	0.404	182.7	217.34	1307.3	29.6	232.9	195.39	1167.62	34015.8	21
14.3	0.42	213.6	250.56	2325.6	28.26	393.43	305.23	123.18	42732.6	21
22.1	0.504	243	293.1	3589.9	22.64	492.8	290.78	173.7	53060	20
28.7	0.555	268.5	307.64	4660	20.1	707.94	161.34	86.08	59942	21.5
35.6	0.642	308.7	346.37	5780.1	16.18	1114.48	44	189.62	77028	21
44.1	0.712	351.3	368.31	7173.8	14.84	1375.48	207.133	95.44	87750	21
51.6	0.8	403.7	415.9	8395.3	12.86	1405.22	97.567	277.16	101196	19
55.5	0.832	444.9	457.61	9052.7	12.24	1347.9	179.84	97.2	110330	21
58.5	0.883	472.8	508.54	9534.5	11.41	1124	174	152.233	117900	21

Catalytic Converter Efficiency For Diesel and Ethanol/DME fuel Emissions:

Table E7: Catalytic Converter Efficiency for Diesel Fuel

1550 rpm Diesel (Standard 3 hole injectors) 210 bar					
Equivalence Ratio	**Bef. Cat Temp (^oC)**	**After Cat Temp (^oC)**	**NO_x (%)**	**THC (%)**	**CO (%)**
0.278	178.8	214	-14.52	6.07	-12.10
0.297	208.7	267	14.87	61.94	94.30
0.367	246.9	311	15.20	64.50	91.91
0.413	285.2	337	23.20	72.48	87.66
0.498	338.1	381	21.86	64.31	85.28
0.568	387.1	426	13.52	65.04	88.04
0.697	437.6	478	1.48	65.76	88.57
0.720	492.4	540	-2.20	75.97	96.44
0.813	532.6	605	-0.25	85.62	97.83
0.974	560.1	785	20.40	89.10	99.04

Catalytic Converter Efficiency For Diesel and Ethanol/DME fuel Emissions continued:

Table E8: Catalytic Converter Efficiency for Ethanol/DME Fuel

1550 rpm Ethanol/DME (Standard 3 hole injectors) 210 bar					
Equivalence Ratio	**Bef. Cat Temp (^oC)**	**After Cat Temp (^oC)**	**NO_x (%)**	**THC (%)**	**CO (%)**
0.375	184.2	194.5	-36.66	76.25	97.39
0.427	184.7	239.7	-85.10	85.30	95.32
0.472	225.4	311.7	21.69	92.59	98.09
0.527	255.8	323.4	14.21	93.23	97.52
0.622	309.7	359.7	23.71	90.88	97.24
0.687	350.3	393	7.93	92.06	96.40
0.781	392.5	439	5.34	92.49	95.47
0.871	476.3	512	0.45	78.84	98.33
0.941	520.4	575	-9.51	81.57	98.28

Catalytic Converter Efficiency For Diesel and Ethanol/DME fuel Emissions continued:

Table E9: Catalytic Converter Efficiency for Ethanol/DME Fuel (4 Hole injector)

1550 rpm Ethanol/DME (Standard 4 hole injectors) 210 bar					
Equivalence Ratio	**Bef. Cat Temp (^oC)**	**After Cat Temp (^oC)**	**NO_x (%)**	**THC (%)**	**CO (%)**
0.404	182.7	217.34	11.50	89.41	75.74
0.420	213.6	250.56	4.37	80.20	96.79
0.504	243	293.1	19.30	77.78	94.20
0.555	268.5	307.64	-3.88	86.10	96.92
0.642	308.7	346.37	3.78	95.69	91.98
0.712	351.3	368.31	11.00	78.89	95.93
0.800	403.7	415.9	2.99	89.09	89.93
0.832	444.9	457.61	-2.38	68.08	97.79
0.883	472.8	508.54	0.97	73.27	98.21

APPENDIX F: Catalytic Converter Efficiency Sample Calculation

Sample Calculation for Test Number ed130306 (Table C12) THC efficiency:

Using equation 3.3.2:

$$\eta_{CAT} = \frac{\left([HC]_{in} - [HC]_{out}\right)}{[HC]_{in}} = 1 - \left(\frac{[HC]_{out}}{[HC]_{in}}\right)$$

where $[HC]_{out}$ = THC specific emissions after the catalytic converter

= 131.2 ppm/kW

and $[HC]_{in}$ = THC specific emissions before the catalytic converter

= 662.9 ppm/kW

Substituting into equation 3.3.2:

$$\eta_{CAT} = 1 - \left(\frac{131.2}{662.9}\right)$$

= 1 – 0.197

Therefore $\eta_{CAT,THC}$ = 80.2 %.